JOURNAL

OF THE

American Society of Naval Engineers.

PUBLISHED QUARTERLY

UNDER THE SUPERVISION OF THE COUNCIL.

VOLUME XXXI.

COUNCIL OF THE SOCIETY:

Rear Admiral H. P. NORTON, U. S. N.
Captain ROBERT STOCKER, (C.C.) U. S. N.
Commander W. W. WHITE, U. S. N. (Retired).
Rear Admiral T. W. KINKAID, U. S. N.
Commander. S. M. ROBINSON, U. S. N.
Capt. of Eng. W. M. PRALL, U. S. C. G.
Commander J. S. EVANS, U. S. N.

WASHINGTON, D. C.
R. BERESFORD, PRINTER, 605 F STREET, N. W.
1919.

50

INDEX TO VOLUME XXXI, 1919.

VOL. XXXI. NO. 1.

JOURNAL
OF THE
AMERICAN SOCIETY
OF
NAVAL ENGINEERS.

PUBLISHED QUARTERLY BY THE SOCIETY, FOR THE ADVANCEMENT OF NAVAL ENGINEERING.

FEBRUARY, 1919.

WASHINGTON, D. C.

R. BERESFORD, PRINTER, 605 F STREET, N. W.

Entered at the Post Office at Washington, D. C., as Second Class Matter.—Acceptance for mailing at special rate of postage provided for in Sec. 1103, Act. of Oct. 3, 1917, authorized on July 3, 1918.

OFFICERS OF THE SOCIETY.

President:

Rear Admiral H. P. NORTON, U. S. Navy.

Secretary-Treasurer:

Lieutenant Commander F. W. STERLING, U. S. Navy (Retired).

Council:

Rear Admiral H. P. NORTON, U. S. Navy.
Rear Admiral T. W. KINKAID, U. S. Navy.
Captain ROBERT STOCKER, (C. C.) U. S. Navy.
Commander S. M. ROBINSON, U. S. Navy.
Commander W. W. WHITE, U. S. Navy (Retired).
Captain of Engineers M. W. TORBET, C. G.
Lieutenant Commander F. W. STERLING, U. S. Navy (Retired).

SUBSCRIPTION RATES.

The subscription price of the JOURNAL, postpaid to the United States and possessions, Canada, Cuba and Mexico, is $5.00. Single copies, $1.30. To other countries in the Postal Union, $5.40. Single copies $1.35. All subscriptions are payable in advance.

Make checks, drafts and postal orders payable to

AMERICAN SOCIETY OF NAVAL ENGINEERS.

Advertising rates will be furnished on application.

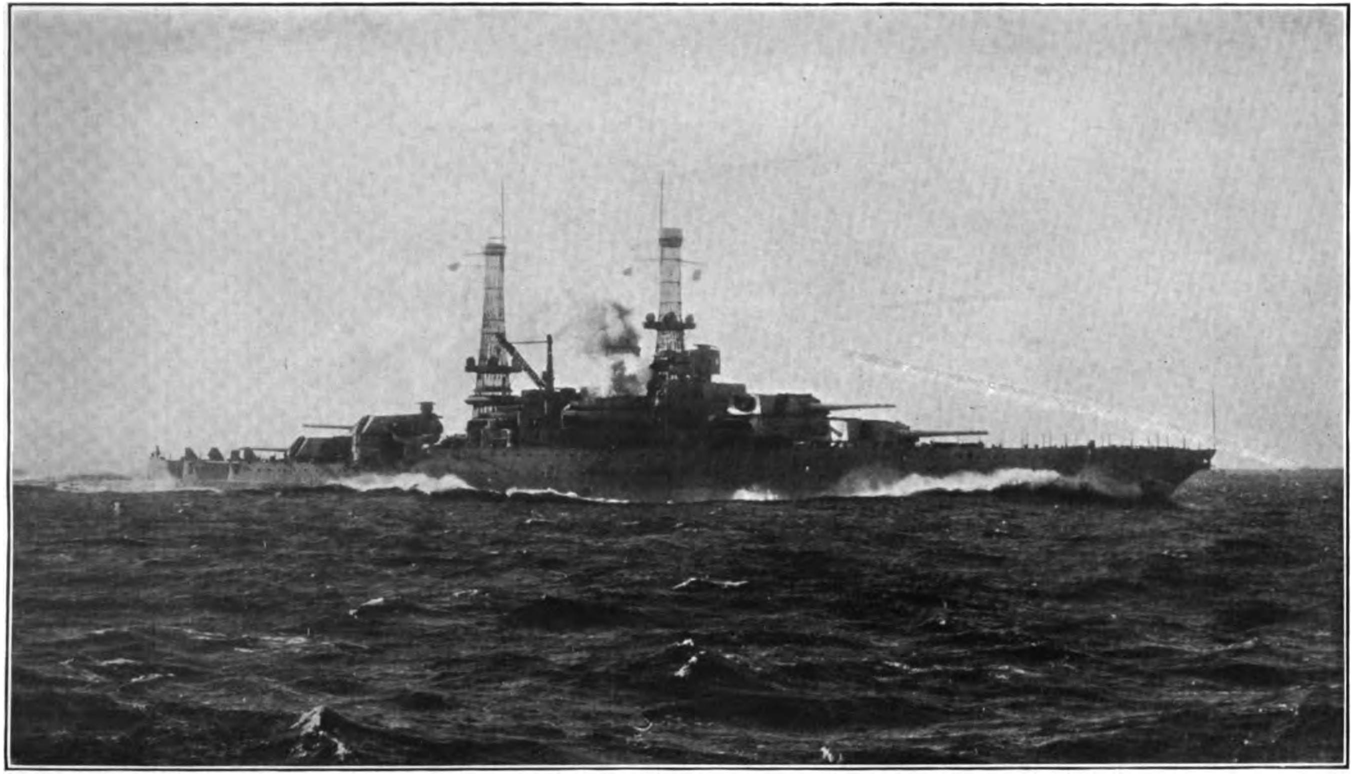
It is earnestly requested that prompt information be given of changes in address, or of failure to receive the JOURNAL.

Address all communications for the Society to

AMERICAN SOCIETY OF NAVAL ENGINEERS,
Navy Department, Washington, D. C.

CONTENTS.

INDEX TO ADVERTISERS.



U. S. S. NEW MEXICO

JOURNAL

OF THE

AMERICAN SOCIETY OF NAVAL ENGINEERS

VOL. XXXI. FEBRUARY, 1919. No. 1.

The Society as a body is not responsible for statements made by individual members.

THE ACHIEVEMENTS OF NAVAL ENGINEERING IN THIS WAR.*

By Lieut. Comdr. William L. Cathcart, U. S. N., R. F., Member.

During these more than four years of war we have heard much in criticism of the Silent Fleets—of the mute guns of that vast Allied Armada waiting tensely, like a crouching lion, for the German High Seas Fleet, which, save for its half-hearted dash at Jutland, never came, until the end in a surrender so ignoble that it sickened the hearts of seamen.

And yet, notwithstanding these criticisms, the ex-Kaiser, his craven commanders on land and sea, and now all Germany know that to those Silent Fleets is due primarily the shattering of their dream of world dominion—a dream whose realization meant to us and to our allies but world despair.

This assertion of the paramount value of sea power involves no detraction from the honor fitly due those magnificent

*This address was delivered before the American Society of Mechanical Engineers and printed in an abridged form in the Journal of that society. It is here published in full with additions.

allied armies, the white crosses of whose dead crowd all Europe. And it minimizes in no way the achievement of the superb troops of the American Expeditionary Force, who, in but half a year's hard fighting against the flower of Germany's armies, have won undying glory for our flag.

But, predominating sea power was the foundation on which all of these victories on the land were based. From the beginning, the British Fleet attained the ultimate object of all sea warfare, in making impotent the enemy's naval strength in surface warships—the only kind that count in the end—and that impotence has had a vital effect on the conduct and success of the war we have waged.

The reason is clear. Every one knows now that an army on the land or a fleet on the sea becomes helpless if its lines of communication with its base are cut. In this conflict, the Western front was the decisive theater of war. And for the allied armies on that front the ultimate bases of supply and reënforcement lay beyond the sea—in Great Britain, her colonies and Dominions; in India and Algiers; and, most important of all, in distant America.

So, the absolutely vital lines of communication of those armies stretched like a vast network across the Seven Seas; and, if Germany's High Seas Fleet had been free to wreak its ruthless will, those lines would have been quickly cut, England would have been isolated, America powerless to aid, and, long before we could have entered it, the war would have ended in bitter tragedy for all mankind that is worth while.

OUR NAVY IN THIS WAR.

While the defeat of the Central Powers is thus due primarily to the early strength and continuous growth of the British fleet, our own Navy has had a far from inconsiderable part in the decisive work of the war's last year, in its service in the Atlantic and the North, White, and Mediterranean Seas, with the ships of the allied navies. On the world's seas, wherever war clouds lowered, our flag flew.

From the very day of America's entry into the war our Navy grew by leaps and bounds, until it became 600,000 men strong and operated about 2,000 armed vessels and transports. On December 1 we had in European waters a total of 338 ships of all classes, carrying 5,000 officers and 70,000 enlisted men, all under command of Vice Admiral Sims. These vessels comprised:

First, a squadron of dreadnaughts in the North Sea; second, cruisers serving with the British in the White Sea and the North Sea, with the British and French in the Atlantic, and with the French, Italian and Japanese in the Mediterranean; third, about a hundred Destroyers operating from several European bases; fourth, a swarm of submarine chasers dashing through British, French and Italian coastal waters; fifth, a flotilla of submarines engaged in hunting enemy boats of their own kind; sixth, a number of small patrol vessels—Coast Guard ships, yachts, seagoing tugs and the like; and, finally, some naval auxiliaries, such as mine layers and repair and hospital ships.

The story of that fleet of ours in those distant, stormy, mist-enshrouded seas, is a very noble one in its record of unceasing toil, of daring and self-sacrifice, of the gallant deeds of the living and the dead. But it cannot be fully told until the censor's ban is lifted.

In his annual report for 1918 the Secretary of the Navy gives averages aggregating 626,000 miles steamed per month as an indication of the work done by all classes of U. S. Naval vessels, engaged in naval duties only, in European war areas. During April, May and June, 1918, our destroyers there escorted 121 troop-ship convoys, consisting of 775 vessels, and during the same period they gave similar guard to 171 merchant convoys, aggregating 1,763 ships.

But the story of the Navy in this war is very far from ending with the record of its ships in European waters. To meet the submarine menace, the United States Naval Cruiser

and Transport Service was organized under command of Vice Admiral Gleaves. When the armistice was granted, this force consisted of 24 cruisers and 42 transports, manned by about 3,000 officers and 42,000 enlisted men of the Navy.

Of the total number of American troops transported to Europe, 46¼ per cent were carried in American ships, and of these all but 2½ per cent were carried in U. S. Naval Transports. In addition, all troops transported in American ships were escorted by U. S. men-of-war. Admiral Gleaves's force thus carried nearly a million troops eastward across 3,000 miles of sea without the loss of a single soldier—an achievement, which, for rapidity of execution and the distance covered, is without parallel in the history of navies. During the last three months of the war, for every minute of the day and night, seven American soldiers, their equipment, and the maintenance for them, arrived in France. As to all this, Admiral Jellicoe says that it was the assistance rendered by the United States Navy which had made the convoy system possible, and that it was this system which had saved the situation.

And finally, we have maintained also a huge Atlantic Fleet, under command of Admiral Mayo, whose vessels have not only guarded our coasts and the near-by ocean lanes, but have formed a great training school from which have come the picked men for European service, the gunners for many hundreds of merchant vessels carrying armed guards, and the crews for Army transports and for the Naval Overseas Transportation Service—that is, the vessels built by the Emergency Fleet Corporation.

When we remember that the maritime area of this country forms so small a part of the whole, the wonder is that, in so brief a period, 600,000 men, coming largely from interior States, could be trained for efficient service afloat in our home and foreign fleets. Sir Eric Geddes justly terms the achievement of the Bureau of Navigation in this work as "an amazing feat."

But what is thus true of one Bureau in this war is equally true of all in the Navy Department—the great shore establishment which stands behind the Navy afloat, and without whose full efficiency effective service by our Fleet would be impossible.

So, what I shall have to tell in a moment of the record made by the Bureau of Steam Engineering should be regarded as but a sample of what all these Bureaus have done.

From the beginning of the war the entire Navy Department has been working, in a clean-cut American way, at maximum efficiency under forced draft.

NAVAL ENGINEERING.

As to my subject this evening, "The Achievement of Naval Engineering in This War," may I say that I shall limit my address to the work of the Bureau of Steam Engineering, and shall be able to handle even that work only in a very general way, since the Bureau's field, which is dynamical engineering largely, covers all steam and internal-combustion engines for the Navy; the bulk of its electrical apparatus for surface ships, submarines and aircraft; the machinery for aircraft; the generation and supply of gas for observation balloons and dirigibles; and, finally, the design and supply of the entire wireless equipment of all shore stations for radio telegraphy in the United States, in our island possessions, and on every vessel, merchant or naval, flying the American flag.

In a broad sense, however, there is much other "naval engineering," although it is static largely, as in the fundamentally important work of the Bureau of Construction and Repair on hulls and their fittings, on stabilizers for ships, and in the Naval Aircraft Factory; and again in the field of the Bureau of Yards and Docks which covers dry docks, buildings and grounds. And, further, the Bureau of Ordnance, in its guns, is but building a special form of internal-combustion engine, and its armor is a product of statical engineering skill.

In fact, in this broad sense, engineering may claim one of the most brilliant achievements of this war, in the battery of 14-inch, high-powered naval guns—having a range, 28 miles, greater than any other guns in France, and mounted on railway cars—with which Rear Admiral Plunkett cut the main artery of German retreat.

So, each of the four technical Bureaus of the Navy Department could tell, if it wished, a noble story of engineering triumphs in this war, although tonight we shall have time for but one.

As to that story, I desire to express my acknowledgment to the Secretary of the Navy, through whose courtesy I am enabled to address this Society tonight, and under whose administration the United States Navy has done such effective work in the greatest war of all time.

May I add also a personal word: after my graduation from Annapolis I was for nearly twenty years in the Navy, but since then I have been long absent from that gallant service. During the Spanish-American War I was on duty in the Bureau of Steam Engineering as one of the assistants to that honored veteran of the Seven Seas and the Arctic ice, the late Admiral Melville.

When, during this war, I came to serve in a similar capacity under his successor, Admiral Griffin, I was amazed at the growth of the operations of the Bureau, both as to their scope and magnitude. From relatively small beginnings in propelling machinery, its field in dynamical engineering had become almost all-embracing, and it had expanded into a huge engineering organization, expending during the last fiscal year more than three quarters of a million dollars a day, having a staff of 137 officers and about 550 employees in Washington, and occupying in the new Navy building there nearly two acres of office space.

So—as an observer simply, and not as one who tells his own accomplishment—I may speak with full freedom of the work

of my associates of years gone by with whom, for a time, it is my honor and my warm pleasure to serve.

THE STUPENDOUS TASK OF THE BUREAU OF STEAM ENGINEERING DURING THIS WAR.

In considering the work of the Bureau of Steam Engineering, let us glance first at the stupendous task which has confronted that Bureau during this war.

When the war began we had a Navy of about 350 ships—commissioned, in reserve, or building. But the fleet grew with amazing rapidity, and, at the close of the fiscal year on June 30 last, the Navy had in service, or soon to be commissioned, a total of 1,959 vessels. This total comprised 570 ships of the regular Navy; 93 drawn from other Government services; 937 converted merchant ships, used as troop transports, naval auxiliaries, and in patrol; and, finally, 359 vessels built by the Emergency Fleet Corporation for the Naval Overseas Transportation Service.

The aggregate horsepower of these vessels is about 6,500,000, which is thirteen times that of our fleet during the Spanish-American War. And, further, it is more than ten times the power developed on both sides of Niagara Falls, and is also equal to about one-sixth of the primary fuel and water power now employed in land service in the United States, not including the locomotives.

But the maintenance of the machinery for this huge power was not all; for at the close of the fiscal year there were under construction 376 combatant and auxiliary vessels and 52 tugs for the Navy. Included in this number is a huge force of Destroyers. When these vessels shall be completed, the present horsepower of the Navy will be increased by 70 per cent, and become 11,000,000.

Neglecting for the time the work involved in designing and building, or inspecting the building of, the engines of ships under construction, let us consider only the duties of the

Bureau with regard to the maintenance of the machinery and apparatus of the nearly 2,000 vessels in service when the war closed.

First, let me note that the Bureau of Steam Engineering is not merely a propelling-engine Bureau, charged only with keeping ships in motion. On the contrary, it is responsible for nearly all of the accessories connected with getting them under way and maintaining them in efficient condition; and, as well, for many of those appurtenances which add to the comfort and health of the crew.

For example, the Bureau must provide an abundant supply of fresh water for, in a battleship, about 1,500 men. It must also furnish machinery and appliances for ice-making and for refrigerating rooms; for electric light throughout the ship and for searchlights; for electric power used in a multitude of ways, from operating a turret to turning an ice-cream freezer; for steam and electric power for cooking, and for steam and electric heating.

The Bureau furnishes also portable electric fans for ventilating spaces not sufficiently served by the main system of the ship; also all kinds of electrical communications, including a complete telephone service; an abundant supply of water for sanitary uses; and the machinery, whether steam or gasoline, for the ship's boats.

For repair work the Bureau provides a machine shop on each vessel with facilities suitable for her ordinary needs. For heavier work it equips the shops of the repair ships which accompany a fleet. It designs and supplies the radio outfit of all ships and shore stations, including those for trans-oceanic service. For submarines it furnishes oil engines for surface cruising, and storage batteries and electric motors for propulsion when submerged. It is responsible for the machinery and wireless equipment of all aircraft, for the generation and supply of gas for observation balloons and dirigibles, and for

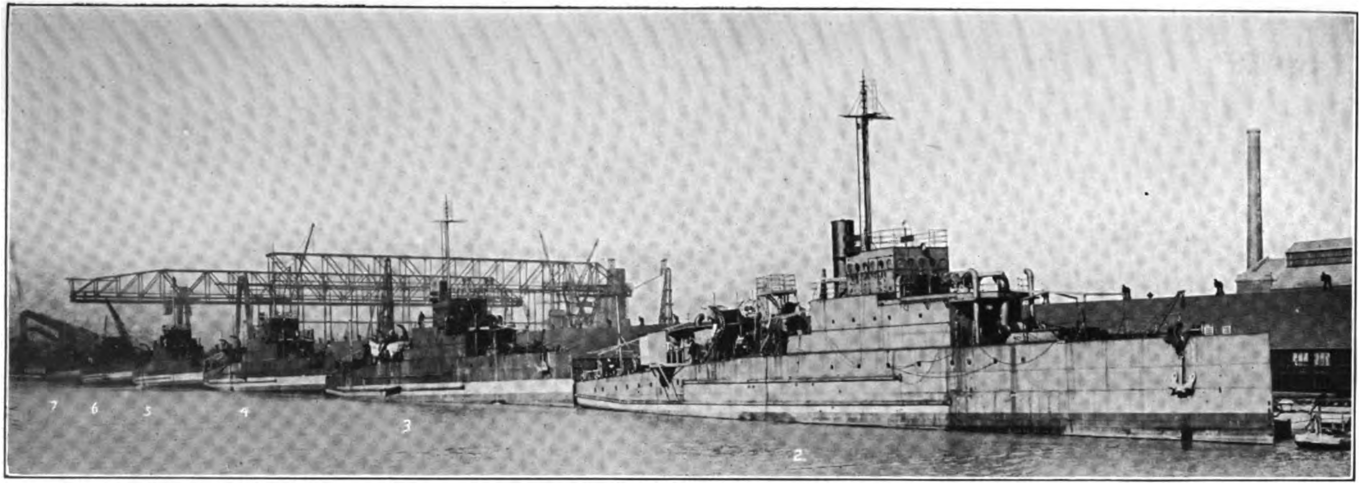


FIG. 1.—THE "EAGLES"—FORD BOATS.

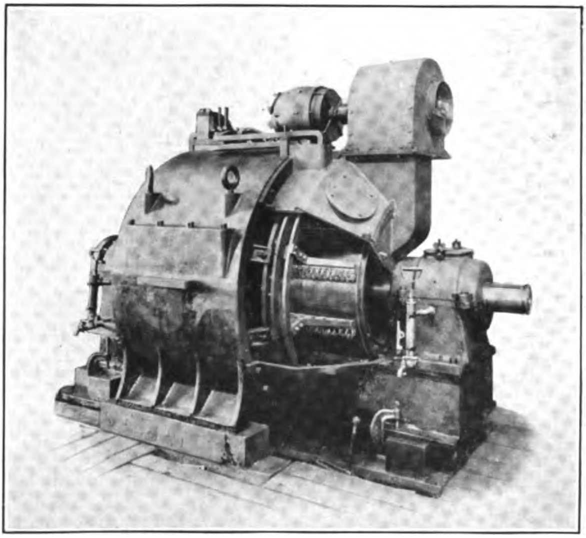


Fig. 2.—Motors for Submarines.

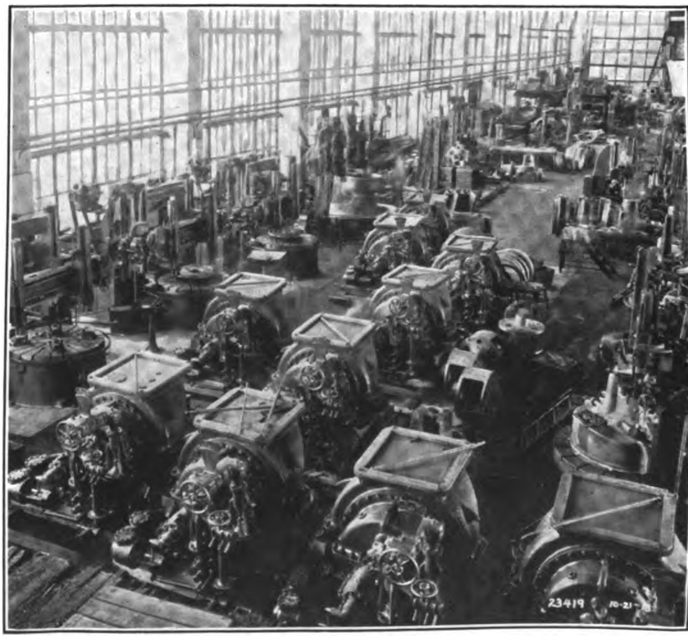


Fig. 3.—Machinery for the "Eagles."

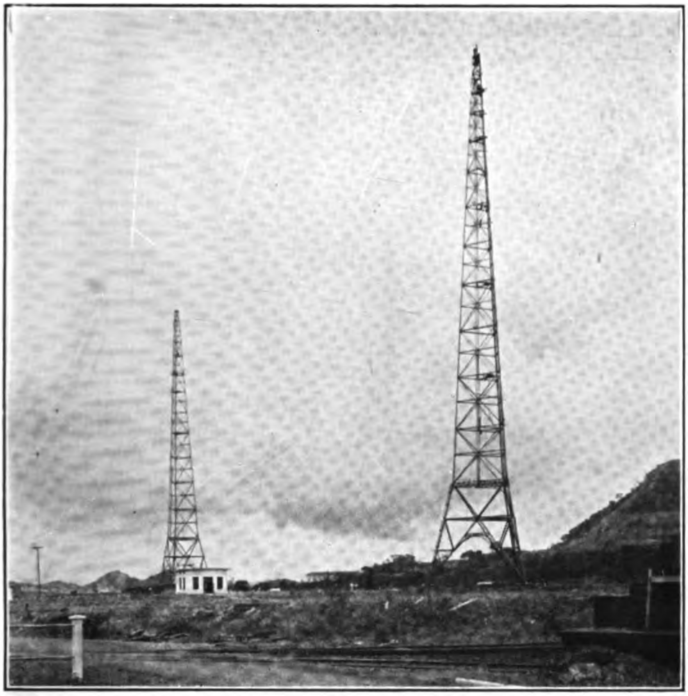


FIG. 4.—U. S. NAVAL RADIO STATION, BALBOA, CANAL ZONE.

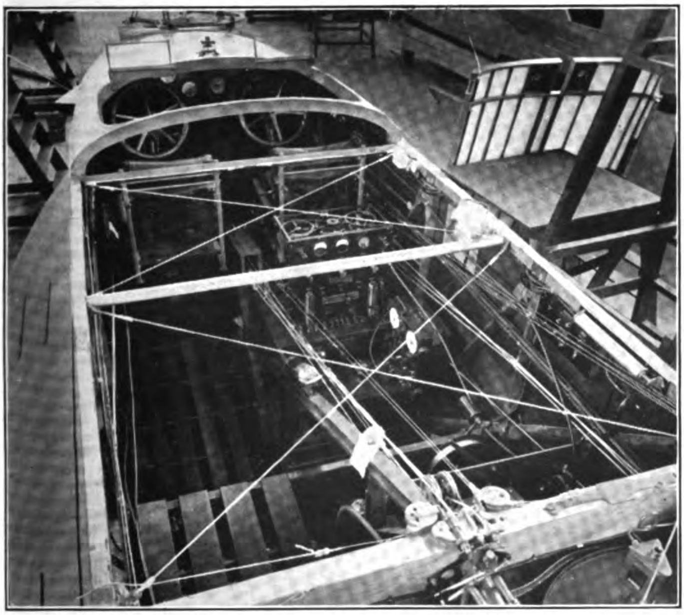


FIG. 5.—AIRPLANE WIRELESS TELEPHONE SET.

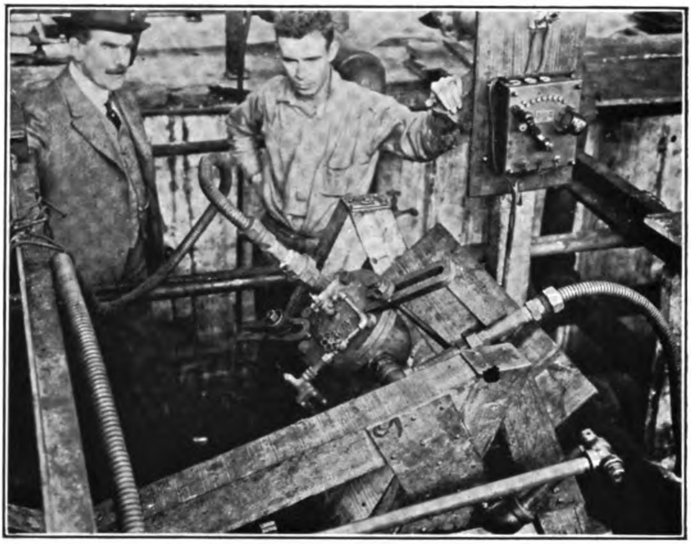


Fig. 6.—Steam Trap Mounted on Rocking Platform to Determine its Behavior on a Vessel in a Seaway.

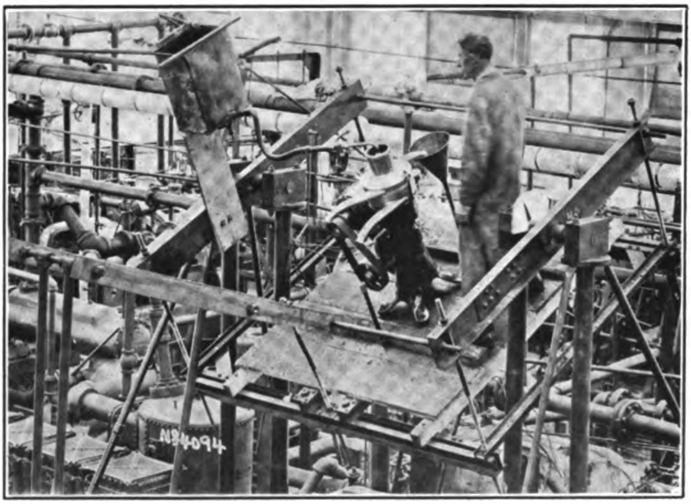


Fig. 7.—Centrifugal Oil Purifier Mounted on Rocking Platform During Development Tests.

FIG. 8.—MICROSCOPIC EXAMINATION OF LARGE METAL OBJECTS. MICROSCOPE AND CAMERA ATTACHMENTS ARRANGED FOR MAKING PHOTOMICROGRAPH OF BROKEN SHAFT COUPLING.

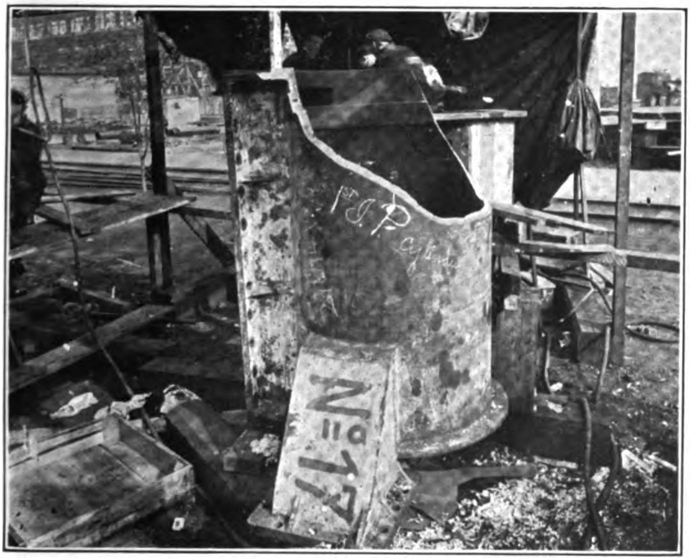


FIG. 9.—1ST I.P. CYLINDER OF S. S. "SANTOS," NAVY YARD, PHILADELPHIA, MAY 3, 1918.

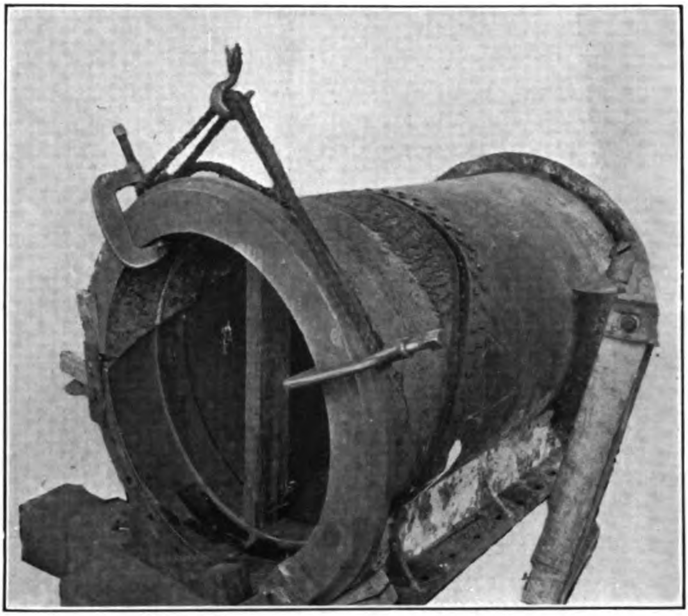


FIG. 10.—1ST I.P. CYLINDER OF S. S. "SANTOS," NAVY YARD, PHILADELPHIA; READY TO WELD, MAY 29, 1918.

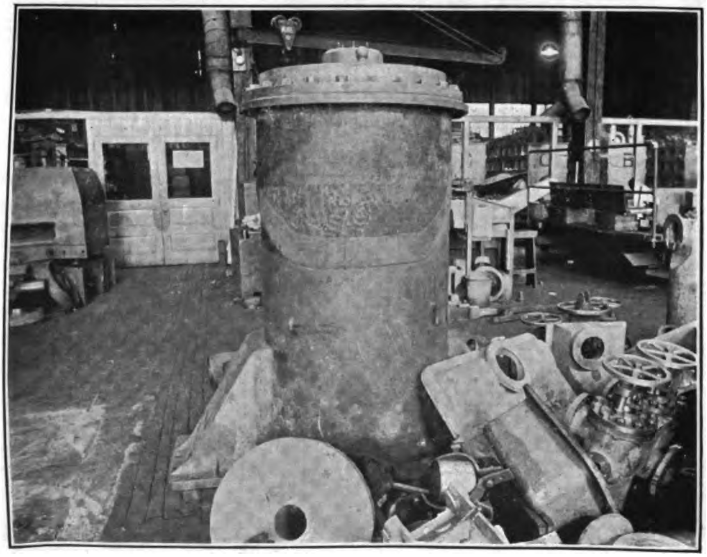


FIG. 11.—1ST I.P. CYLINDER OF S. S. "SANTOS," NAVY YARD, PHILADELPHIA; FINISHED, AUGUST 15, 1918.

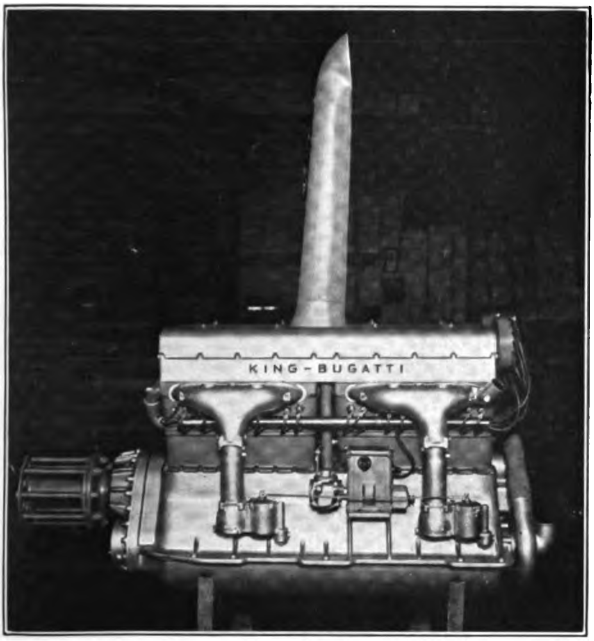


Fig. 12.—King-Bugatti Airplane Engine, 16 Cylinders.

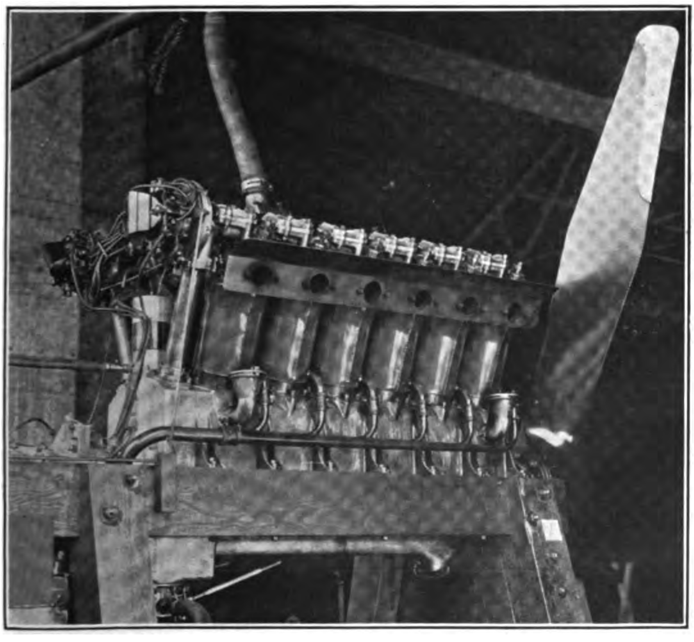


Fig. 13.—Liberty Motor, 12 Cylinder, 400 H.P., Standard U. S. Aeronautical Engine.

FIG. 16—STACK INSTALLED ON DESTROYER.

FIG. 15.—WHIPPING STACK TO DESTROYER.

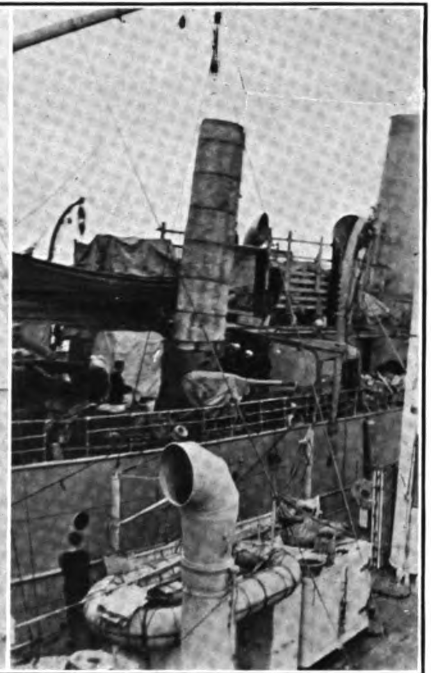


FIG. 14.—COMPLETED STACK ON U. S. S. "DIXIE."

SMOKE PIPE MANUFACTURED ON U. S. S. "DIXIE" AND INSTALLED ON U. S. S. "AMMEN" IN TOTAL PERIOD OF SIX DAYS.

the detection devices used against enemy airplanes and submarines.

When we entered the war there was naturally a tremendous increase in the routine work of the Bureau. For the Battleship Fleet, although always ready for normal duty, needed finishing touches to fit it for possible service abroad. Also the Reserve Fleet had to be put in full commission, and the Destroyers and their accompanying Repair Ships had to be prepared and fitted out for their work in European waters.

Further, the converted merchant vessels taken over by the Government had to be repaired, altered and fitted out for use as naval auxiliaries and transports; the many merchant ships carrying armed guards had to be provided with electrical equipment—call bells, telephones, and the like—for the effective control of gun fire; and the unprecedented work of repairing the damaged machinery of the German merchant vessels we had seized had to be done at express speed. To all this there was added almost immediately the construction of a large number of submarine chasers, and later, of a huge force of destroyers and mine layers.

From this hasty outline some idea may be gained of the scope and magnitude of the work of the Bureau of Steam Engineering during the war. That, notwithstanding the enormous expansion of our Fleet in this period, it has been possible to carry on this work successfully is due to the efficiency of the prior organization of the Bureau. The Engineer-in-Chief did not create a new system, but simply expanded the one already existing, which is the development of one inaugurated thirty years ago by a former engineer officer of the Navy, Mr. Asa M. Mattice, a distinguished member of this Society.

This also may be said as to such advances as the electric drive for battleships, the exceedingly compact arrangement of machinery of 28,000 horsepower in the latest destroyers, the noiseless and vibrationless engines of the Ford boats, the progress in radio equipment and in anti-aircraft and anti-

submarine devices and other minor matters. In all these the Bureau has been successful virtually at its first attempt. And this success has not been haphazard, but was the result of careful, prior investigation and planning by an efficient organization.

In its work the Bureau of Steam Engineering has spent huge sums during this war—more than 283 million dollars, for example, during the last fiscal year. Tonight, I hope to give you, as citizens and tax-payers, a fair amount of evidence that your money has been well spent, not only by this Bureau but by all others in the Navy Department.

ORGANIZATION OF THE BUREAU OF STEAM ENGINEERING.

For clearness in discussing the work of the Bureau of Steam Engineering let me give a brief outline of its organization and principal activities.

The Engineer-in-Chief determines all questions of Bureau policy, authorizes all expenditures, and decides all important—and many minor—questions of detail.

The Assistant to the Bureau, Captain Oscar W. Koester, U. S. Navy (retired), handles routine business in technical and administrative matters; investigates all work and expenditures in connection with the organization, building, equipment and maintenance of the shops of the Machinery Division at Navy Yards and Repair Bases, and, when necessary, inspects and reports on the condition of the Bureau's work at private plants.

The staff of the Bureau is separated into ten divisions—one clerical, headed by the Chief Clerk; and the remainder, technical. The duties and titles of the nine latter divisions are:

Design, Electrical, Repairs, Radio Telegraphy, Inspection, Supply, Fuel and Personnel, Aeronautics, and Logs and Records.

Each of these divisions is in charge of an officer who is an expert of high rank in his especial field. The muster roll of

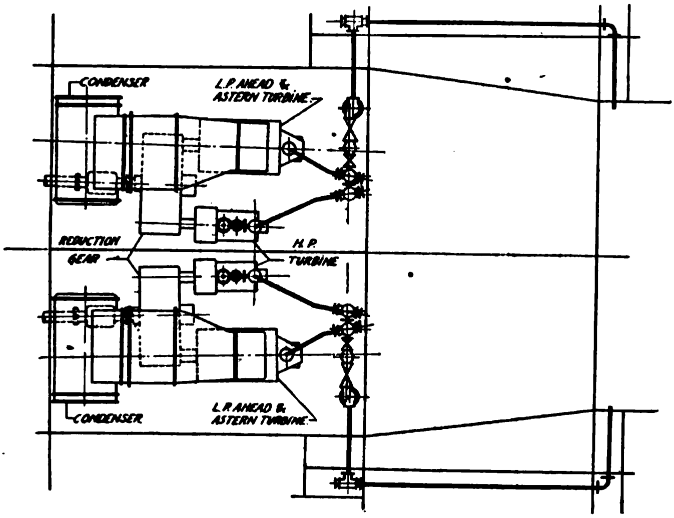


FIG. 17.—BATTLESHIP MACHINERY ARRANGEMENT; DIAGRAMMATIC PLAN OF TURBINE DRIVE WITH REDUCTION GEAR.

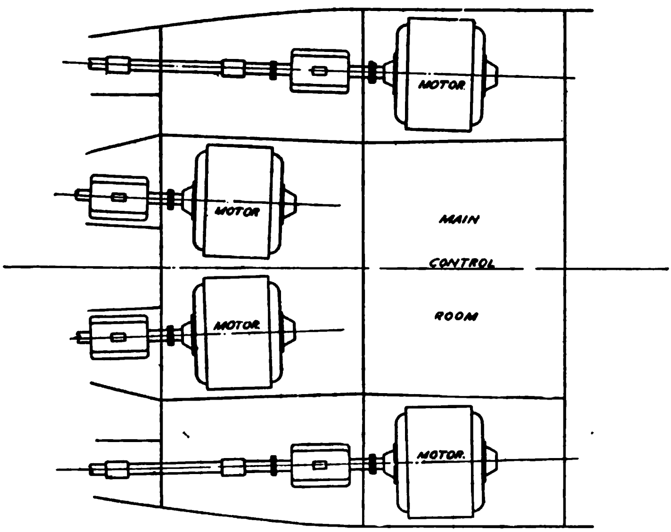


FIG. 18.—BATTLESHIP MACHINERY ARRANGEMENT; DIAGRAMMATIC PLAN OF ELECTRIC DRIVE SHOWING ARRANGEMENT OF MOTORS.

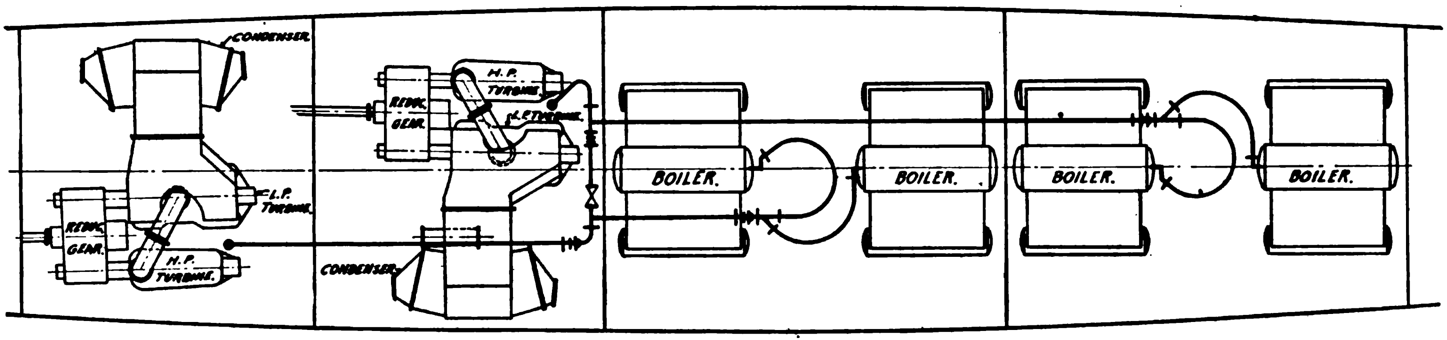


FIG. 19.—DESTROYER MACHINERY ARRANGEMENT; DIAGRAMMATIC PLANE OF TURBINE DRIVE WITH REDUCTION GEARS.

the Bureau's divisional heads is in this respect a roll of honor. While the work of the Bureau is thus divided, Admiral Griffin keeps in close touch with it all. His guiding hand is felt, in greater or less degree, in every detail.

Division of Design.—The Division of Design is in charge of Rear Admiral Charles W. Dyson, U. S. Navy, an officer who has long been widely known for his success in the design of naval machinery, and especially of propellers—that marine mystery of many weary years.

The duties of this Division cover a broad field. As to new vessels, for example, it recommends the type of machinery to be adopted; estimates the power required for propulsion; makes the general design of the machinery selected by the Engineer-in-Chief; draws the contract-plans and specifications; investigates the bids and alternative plans submitted by contractors, and makes recommendations; criticises the detailed plans forwarded by contractors; prepares directions for the trial of the machinery; and finally makes recommendations as to payments on the contracts.

During the last two years of the war this comprehensive work was done for vessels of all classes—built, building, or contracted for—aggregating 9,501,440 horsepower, as is shown by the accompanying table.

Destroyers	6,578,000
Battle Cruisers	1,080,000
Battleships	480,000
Scout Cruisers	630,000
200′ Patrol Boats	280,000
Mine Sweepers	75,600
Sea-going Tugs	48,600
Submarines	41,640
EFC Oil Tankers	31,800
Harbor Tugs	12,000
Fuel Ships	10,400

Motor Tugs	2,800
Ammunition Ship	10,600
Oil and Water Barges	2,400
Gunboat	1,600
Total H. P.	9,285,440
110′ Patrol Boats	216,000
Grand Total	9,501,440

In addition to the work on new vessels the Division passes on all alterations to the existing machinery of naval vessels and those of the Naval Overseas Transportation Service. It also specifies the materials and their characteristics for marine engineering construction, and has general supervision of all tests and experiments relating to marine propulsion at the Engineering Experiment Station at Annapolis and the Fuel-Oil Testing Plant at the Philadelphia Navy Yard. Similarly, it has general supervision of the Inspection Force at building yards, a work in which the Bureau has at this time about 60 officers engaged.

Electrical Division.—When we entered the war the Electrical Division was in charge of Commander Guy W. S. Castle, U. S. Navy, who has been succeeded recently by Captain C. E. Courtney, U. S. Navy.

This Division supplies all the electric power, the wiring for all purposes, and, roughly, about 75 per cent of all electrical machinery and apparatus on naval vessels. Its field is therefore a most important one, extending to communication systems for the control of gun fire, the electrical signals for day and night use which are so necessary for the handling of ships in company at sea and in making recognition signals, searchlights for protection against Destroyer attack at night, the motors and storage batteries for the propulsion of submarines when submerged, anti-submarine and anti-aircraft devices, all interior communication systems, the multitude of other appli-

ances in which electricity serves on shipboard, and the electrical equipment of the Machinery Division at Navy Yards. In this work the Bureau has expended about $20,000,000 during the last fiscal year.

When the United States entered the war the Electrical Division was confronted by a huge task, since Congress had failed to appropriate sufficient funds for an adequate amount of spare material and there was little in stock at Navy Yards. For example, the amount of wire in store was almost negligible, while day by day the number of ships acquired in various ways grew with amazing rapidity.

But, six months before we became a belligerent, Commander Castle, by direction of the Engineer-in-Chief, had made a comprehensive survey of all probable requirements in the event of war, and hence he was ready to act promptly and effectively. Manufacturers everywhere were patriotically eager to aid, and so the situation was saved.

As an instance of this: Before the war wire was supplied by only two or three firms. When enormous quantities were thus needed immediately, fifteen of the largest wire manufacturers were summoned to Washington, and contracts were made with them which virtually gave the Bureau control of the wire output of this country.

In original work Commander Castle supervised the development of certain anti-submarine and anti-aircraft devices which are of a confidential nature. These submarine detectors have been made in quantity, not only for our own vessels, but for the British Admiralty and the French Ministry of Marine. For the anti-aircraft investigation Dr. G. W. Stewart, of the University of Iowa, made for the Bureau a comprehensive study at Pensacola, Florida, of all the noises made by the engine and wings of an airplane in flight. From his results and in coöperation with the Signal Corps of the Army, a scheme of sound-ranging on enemy airplanes was devised, and the necessary instruments were designed and made.

Unquestionably the most important electrical accomplishments of the Bureau during the war have been the remodeling and improvement of the wiring for gun-fire control systems in all ships from dreadnaughts to gunboats, the installation of communication systems for the service of the guns on transports and ships carrying armed guards, and the provision of recognition signals on all vessels acquired or operated by the Navy.

Not the least part of the electrical work of the Bureau is concerned with storage batteries for submarines. These serve not only for propulsion, but for nearly all auxiliary purposes, such as pumping, lighting, heating and cooling. The continuance of the war brought great growth in the number of our submarines, with consequent increases in their operations and the spare equipment that it was necessary to carry. All of our submarine batteries are of the lead-acid type. In some cases the expedient was adopted of reducing the thickness of the plates in order to increase the exposed surface. While the ultimate effect of this is to reduce the life of the battery, the military advantage gained in thus obtaining one of greater power for the same weight, was considered sufficient to offset the disadvantage of short life.

Owing to the magnitude of the work of the Electrical Division and the fact that so many regular officers of the Navy who were qualified for this work were ordered to sea duty, the Bureau succeeded in having one hundred graduates in electrical engineering enrolled in the Naval Reserve and given a short course in training for naval work. These young officers have been serving very creditably in the Fleet.

Supply Division.—The Supply Division is in charge of Lieutenant Commander Charles K. Mallory, U. S. Navy (retired), a member of this Society, who, on the mobilization of the Navy for war service, was assigned to this duty as the relief of an officer on the active list.

Matters relating to engineering materials and supplies—

under cognizance of the Bureau of Steam Engineering—for vessels of the Navy and for Navy Yards, are handled through this Division. The specifications for the material are drawn by the technical Division concerned, and all subsequent questions of a technical nature are referred to that Division for decision. Hence, the Supply Division is, in some respects, a sort of center of business activity for the Bureau in its contact with bidders and contractors. During our first year in the war it passed upon materials aggregating $100,000,000 in value, and this was for miscellaneous materials only, and did not include new construction work. The actual purchase is made by another Bureau, which refers to the Bureau of Steam Engineering all bids for determination as to the one that is most advantageous.

Under peace conditions—when contracts for the construction of vessels as well as the purchase of supplies were made on a fixed price basis, with a penalty for failure to complete or deliver on time—the duties of the Division consisted primarily of passing upon requisitions for materials as to their quantity, quality and suitability for the purpose intended, and of analyses of the proposals received from bidders with recommendations as to the award of contracts. In addition to this service the Division was charged with the duty of providing suitable allowance lists, covering spare parts and material for the outfitting and operation of vessels of the Fleet.

Upon our entry into the war it soon became apparent that the question of obtaining material and supplies—both for the direct use of the Navy and for those having contracts for new construction—would be of increasing importance and difficulty. Owing to the unsettled conditions as to materials, supply and labor, it became necessary to abandon the penalty for failure to complete contracts on time, and in many cases contracts had to be changed from a fixed price basis to one which would insure the receipt by the contractors of a reasonable profit.

Many new Government activities, as well as others of a private nature closely associated therewith, came into existence, requiring material in excess of the country's output. The War Industries Board and other Commissions and Committees, with which we are all more or less familiar, were formed to control the resources of the country and to direct material into channels where it was most needed. All of this meant that the Navy's engineering requirements had to be closely studied in order to insure delivery of material when and where required, and to avoid the making of unnecessary demands at the expense of other essential war activities.

Owing to the change in the form of contracts, various questions were continually arising as to which the Bureau was called upon for an opinion or a recommendation. The handling of such situations was centered in Washington, the Supply Division being expanded as occasion required to take care of the increasing volume and variety of work. A follow-up system for all requests for material was also instituted, by means of which the exact status was shown at any time up to the actual completion of contracts.

Under peace conditions, after the placing of a contract for engineering material, either for the direct use of the Navy or for Government contractors, it was inspected as to quality before delivery by the various Naval Engineering Inspectors throughout the country. Also, it was unnecessary in those days to give much attention to the matter of the ability of the contractor to deliver as required. This latter consideration, however, became more and more important in war time, and. to take care of the situation, a number of business men of ability, who were willing to devote their time to Government needs, were enrolled as officers in the Naval Reserve Force and assigned to production work in the various Inspection Districts.

The duty of these officers was to follow closely every contract; to know whether it would be filled as required; and, if

not, why—so that any necessary action could be taken promptly. In addition to the regular Inspecting Force of the district, who aided in this work, enlisted men were also provided by the various Naval Districts. The work of these production officers was of inestimable value in obtaining material as required.

The Supply Division has also made a revision of the engineering allowance lists for all regular Navy vessels, to insure their being on a proper war footing in this respect. It will be seen that, owing to the variety and volume of its work, this Division has had its hands full since our entry into the war.

Radio Division.—Commander S. C. Hooper, U. S. Navy, is in charge of the Division of Radio Telegraphy. The activities of the Bureau of Steam Engineering in this respect cover almost a world-wide field. All matters relating to radio equipment—except the actual operation by the radio personnel—on vessels operated by the Navy, on all vessels of the Army and other branches of the Government, including the United States Shipping Board, and on merchant ships requisitioned by that Board, are directed by the Bureau. Virtually, therefore, the radio installations on all vessels flying the American flag is, in these respects, under the direction of the Bureau of Steam Engineering. The number of these installations now exceeds 4,000.

The Bureau's direction extends similarly to 50 naval radio coastal stations and 75 commercial coastal stations in this country; to others in the West Indies and the Canal Zone, in our island possessions and Alaska and Vladivostok in the Pacific; and, finally, to one now building for us at Bordeaux in France, which will be the most powerful wireless station in the world.

Our Atlantic, Pacific and Gulf Coasts, and the shores of the Great Lakes, are covered by strategically located low-power and semi-high power radio stations. Washington is in touch also with our possessions in the West Indies by means of the

low-power stations at St. Thomas, and Santa Cruz, San Domingo City, and Haiti in the Republic of Haiti, by way of the medium-power stations at San Juan, Porto Rico, and Guantanamo, Cuba. Also, the high-power station at El Cayey, Porto Rico, now nearing completion, will ensure direct communication between all points in the West Indies and vessels in southern waters, and meet the demand for such facilities with South America.

Communication between the Canal Zone and Washington is maintained by the medium-power stations at Colon and the high-power station at Darien, the latter working directly with Arlington at all times. In the Pacific we have high-power stations at Cavite in the Philippines, Guam southeast of it, and Tutuila in Samoa, which maintain communication with Washington by way of Pearl Harbor, Hawaii, and San Francisco or San Diego. Arlington communicates with Alaska by way of various medium and high-power naval radio stations located in Alaska and the high-power station at San Francisco or San Diego.

Communication between Washington and Russia will be made through the establishment of the Vladivostok station, now nearing completion. Our communications with China are through Cavite, and our Asiatic Fleet to the United States Naval Radio Station at Peking.

During the last fiscal year four of the high-power radio stations on the Atlantic Coast—Sayville, Tuckerton, New Brunswick and Marion—have been developed into efficient transmitting stations capable of continuous radio communication with Europe. And, further, the Annapolis transmitting station, the most powerful in the United States, was built and commissioned. This station and the four just noted would provide uninterrupted communication with our forces in France if all submarine cables were cut.

The Bureau has also conducted a large amount of investigation and development of radio transmitting and receiving

apparatus for aircraft. It has equipped 90 airplanes in the United States with satisfactory apparatus, has shipped 60 outfits to France and Great Britain, and has contracted for about 3,000 more.

Radio telegraphic communication from aircraft in flight to stations on land is now possible at a distance of 200 miles. Similar communication from land stations to flying aircraft is practical up to a distance of 50 miles, and communication from aircraft resting on the water to points on shore can be effected at a maximum distance of 40 miles.

The development of the radio telephone for use on aircraft has progressed to such a stage that it is now possible to communicate by this means from aircraft in flight to stations on land at a maximum distance of 60 miles. The converse of this—telephone communication from land stations to flying aircraft—is practicable at a distance of 15 miles.

Results such as those I have noted show that the efforts of the Bureau—in coöperation, in some cases, with outside organizations—have resulted in the development of aircraft radio equipment of remarkable efficiency. In fact, a direct comparison of American and foreign systems shows that the Bureau's apparatus accomplishes with one set what, with European equipment, can be done only by two separate sets and an additional hand-driven generator.

Inspection Division.—The Inspection Division is in charge of Commander M. A. Anderson, U. S. Navy (retired). The inspection of material for new construction, spare parts, and repairs is one of the most important branches of the work of the Bureau of Steam Engineering, not only in its own field but for other Bureaus. During the last fiscal year this Division inspected, for its own Bureau and six other Bureaus of the Navy Department, nearly eight million pounds of engineering and other material. This inspection was conducted by a force of 306 naval officers and civilian assistants who visited more than 2,000 manufacturing establishments in this work.

It should be noted that the weight measurement is not a trustworthy gage of the amount of work done in inspecting material. For example, tons of ingots, pig iron, or forging billets may be examined and accepted by an inspector in a single day, while several days may be necessary to inspect certain aircraft material or delicate machinery weighing but a few pounds.

Fuel and Personnel.—The Fuel and Personnel Division is in charge of Commander H. A. Stuart, U. S. Navy. During the greater part of the war, it was headed by Commander N. H. Wright, U. S. Navy.

Fuel oil, gasoline and lubricating oils are vital essentials of modern navies, hence the work of this Division is of fundamental importance, especially as the Navy is now hard pressed in its effort to retain the naval oil reserves in California. However, all of the Navy's new ships are oil burners, since, during their lifetime, fuel oil for them will not fail.

This Division covers a broad field. During the last fiscal year it assisted in extending the coaling facilities in our ports and in making a comprehensive survey of the low-volatile steaming coals with the view of locating mines capable of furnishing suitable fuel for the Overseas Transportation Service. It modified the naval specifications to permit the use, for a time, of a considerable percentage of Mexican distillate, and made preliminary investigations for establishing specifications for heavy fuel oil for transports and cargo carriers. Further, it coöperated with various Government agencies in the adoption of standard specifications for aviation gasoline.

The Division has also coöperated with the Bureau of Navigation in the effort to obtain officers qualified for engineering duty in the regular Navy and in the Naval Reserve. Virtually all of the nearly 2,000 ships built and to be built by the Emergency Fleet Corporation are to be operated by the Navy, a service which will require a total of about 25,000 deck and engineer officers—and that is another matter in which the Navy is hard pressed.

Division of Repairs.—The Division of Repairs is in charge of Commander W. A. Smead, U. S. Navy, who has been for some months in Europe. In his absence it is headed by Lieutenant Commander Bruce.

Next to importance to building a ship is keeping it in repair. Lame ducks cannot fight. During the war this Division has had its hands full with the maintenance of, and the repair work for, the machinery of the Fleet and the vessels operated by the Navy—a total of nearly 2,000 by the end of last June. However, it measured up fully to all requirements. No vessel of importance has been laid up for repairs. In addition to this it has handled all matters connected with the taking over and fitting out of vessels acquired or to be operated by the Navy.

Since we had more than 300 vessels in European waters, the problem of their repair and maintenance was not simple. Spare parts had to be ready and matériel needs anticipated. To this end a stock of the most important engineering material was provided by this Division—some in this country, but the greater portion at the foreign bases—and its location and use so thoroughly systematized that it could be shipped upon cable advices.

It is noteworthy in this connection that when one of the Destroyers was so badly damaged by a submarine that it was with difficulty that she was able to make port, spare shafting and propellers were ready for her long before their installation was necessary. In another case, where it was seen that the boilers of a Destroyer needed retubing and the time required would have put her out of service for too long a time, new boilers were built and, when ready, were installed with a surprisingly short period of inactivity.

For this work of repair and supply the Navy established a total of five major bases—three in France, one in Great Britain, and one at Gibraltar. There are also two minor bases in or near the Mediterranean. The facilities of these bases were reënforced by those of six Repair Ships, general plans of

whose shops are shown in the views. These vessels proved to be of the greatest value to a fleet in a war area, and their work was frequently commented upon, and very favorably, by both Vice Admiral Sims and the officers of foreign services. Without these ships, our Destroyers could not have maintained the wonderful record they established of being always ready. In this respect our Navy leads all others.

Our Navy had an average of approximately 50 Destroyers serving for 19 months in European waters. One Repair Ship there re-tubed two Destroyer boilers (Normand) in nine days, the boilers being in place. The services of these Repair Ships were so effective that only $1,100,000 were spent on this whole Destroyer Force for machinery repairs made by other agencies during this period, and this total covers several cases in which there were very serious damages by torpedoes or collisions.

As illustrating the resource of the Repair Ships, the retubing of boilers of a Destroyer was undertaken without interruption of her regular patrol and convoy duty by working on one side of the boiler during her 5-day overhaul period and completing the other side during the following period, and so on until all were finished, the vessel meanwhile being capable of 25 to 28 knots speed.

It is doubtful whether such a performance would have been possible under other direction than that of Vice Admiral Sims, who insisted that Destroyers must have a regular overhaul period uninterrupted by any other duty or consideration.

Division of Aeronautics.—The Division of Aeronautics is in the charge of Commander A. K. Atkins, U. S. Navy. The work of this Division covers aircraft machinery generally, engineering design for this purpose, and the generation and supply of gas for observation balloons and dirigibles.

The marked military advantage to be gained by having but one type of airplane motor prompted the Navy Department to adopt the motor used by the Army. All Liberty motors for the Navy are, therefore, obtained through the Aircraft Pro-

duction Board. Orders have been placed for more than 4,000 of these Liberty engines, delivery to be completed by January 1, 1919. Of this number about 1,300 were received by the end of the last fiscal year, and distributed to various naval air stations in this country and abroad. There have also been purchased about 500 Hispano-Suiza engines for use at naval air stations in this country.

At various times the Bureau of Steam Engineering has tested and used a total of eleven airplane engines as follows:

Motor.	*H.P.*	*Weight, Lbs.*	*Remarks.*
Liberty, 12 cyl................	380	852	Low comp. water cooled.
Geared Liberty, 12 cyl........	450	972	V-45°; water cooled.
Kirkham, 12 cyl...............	400	670	V-60°; water cooled.
Dusenberg, 16 cyl............	800	1,617	V-60°; water cooled.
Bugatti, 16 cyl................	475	1,170	Cyls. in parallel rows of eight; crank shafts geared together.
Kessler supercharger..........	600	600	6-cyl., vertical.
Hall-Scott, Liberty Six.......	212	490	Vertical, water cooled.
Union Engineer, 6 cyl.........	125	443	Vertical.
Hispano-Suiza, 8 cyl..........	150	440	V-90°; water cooled.
Curtiss, OXX-6, 8 cyl.........	100	395	V-90°; water cooled.
Curtiss, V-2, 8 cyl...........	200	690	V-90°; water cooled.

Among the more important of these motors are: the Liberty, 12 cylinder; the Geared Liberty, 12 cylinder; the Hispano-Suiza; the Union Engine; and the Kirkham Motor.

In general engineering design the Bureau has done much work, especially with regard to the application of the Liberty Motor to seaplanes. This work covered starters and priming devices; the development of a flow meter, pneumatic-spark arc-throttle controls, and an oil-cooling system.

One of the very important duties devolving on the Bureau of Steam Engineering is the equipment and maintenance of stations for the generation and supply of a non-inflammable gas for use in observation balloons and dirigibles. Somewhere in the United States there is a plant for this purpose, on which the Bureau was authorized to expend $3,000,000. A number

of stations have been established and a full equipment of gas-containers provided, so that any calls may be promptly met. This gas has been shipped in quantity to the British and French. For this balloon-gas service it has been necessary to give suitable training to a number of young officers and to assign them to stations at home and abroad.

Division of Logs and Records.—The Division of Logs and Records is in charge of Commander W. W. White, U. S. Navy (retired). In the effect of its work on marine engineering design and progress, the field of this Division is of great importance.

Its main functions are: the checking up and review of the engine-room logs of naval vessels; the compilation and issuance to the seagoing personnel and naval training schools of Bureau pamphlets relating to the care and management of machinery on shipboard; the preparation and issuance to the naval service of the "Confidential Bulletins of Engineering Information;" and the filing and indexing of all important data regarding engineering matters, with especial reference to naval vessels.

It will be seen that the work of Commander White—an officer of long experience in this duty—is of marked importance, not only to the Navy, but to the engineering profession.

NAVY YARDS AND SHORE STATIONS.

The efficiency of the Machinery Divisions at Navy Yards and Shore Stations has a most important effect on the general success of the Bureau's work, in keeping ready for service the machinery of the Fleet and of other vessels operated by the Navy, and also in building machinery for new vessels.

During the last fiscal year, in nine Navy Yards and Naval Stations in the United States and our insular possessions, there were employed on Bureau work exclusively about 175 commissioned and warrant officers and 9,700 mechanics. In four other Navy Yards, under Industrial Managers who direct the

work for all *matériel* Bureaus, about 110 commissioned and warrant officers and 11,500 mechanics were engaged in these activities.

An adequate resumé of the accomplishment of these large forces during this war would take far more time than we have available this evening. However, as an example, I may give a few particulars as to the immense amount of work carried on at the New York Navy Yard under the direction of Rear Admiral George E. Burd, U. S. Navy, Industrial Manager. Since April 6, 1917, Admiral Burd and the officers serving with him have had charge of the conversion and repair, in all departments—including the mounting of batteries in many cases—of a total of 723 vessels of all classes. Much of this work was under cognizance of the Bureau of Steam Engineering.

In addition to this, the propelling machinery, in whole or in part, of three battleships has been building there, and considerable other new construction work of a minor character. Further, there has been a great deal of electrical and radio work for vessels of all classes. The number of men employed—excluding the clerical, drafting, and sub-inspector force—was about 4,900.

In connection with the Navy Yards, a very large amount of work has been done by private establishments on naval vessels and transports. And in the prosecution of this work the closest coöperation prevailed between the Commandants of the various Naval Districts and the Commandants of Navy Yards and the Bureau of Steam Engineering, and also with Vice Admiral Gleaves, in command of the troop transport service. Without the team work that thus prevailed and the harmony of interest ever uppermost, many a sailing would have been delayed.

NAVAL EXPERIMENT STATION AT ANNAPOLIS.

The Naval Experiment Station at Annapolis does much of the marine engineering research and experimental work of the

Bureau. The Head of this Station is Rear Admiral T. W. Kinkaid, U. S. Navy.

The services of this Station are of the greatest value, not only in the designing work of the Bureau, but in the maintenance of the machinery of the Fleet. Its chief function is the prosecution of research work on, and making tests of, appliances, materials, and methods which relate to the propelling plants of naval vessels, such as forced-draft blowers, pumping machinery, feed-water heaters, coolers, refrigerating machinery, steam turbines, condensers, evaporators, boilers, coal and oil fuels, packings, and so on. The Station's activities also include much work in the metallography of steel and other metals.

FUEL-OIL TESTING PLANT.

The Fuel-Oil Testing Plant at the Philadelphia Navy Yard has also done a very considerable amount of valuable work during the war. Its present head is Lieutenant A. M. Penn, U. S. Navy. Its field covers burners, air registers, the refractory linings for furnaces, furnace insulation, and other similar matters.

During the last fiscal year, in order to obtain comparative data of boilers of various types, a test of the White-Forster boiler was conducted at this plant. The results show that express type boilers of this kind, having a large amount of heating surface and a thick tube bank, are even more efficient, at from low to moderately high rates of combustion, than the straight-tube boilers now used so extensively in our Navy, but that they are slightly less efficient at very high rates. The express type has advantages also in the saving of weight, original cost, and cost of upkeep.

SOME RECENT DESIGNS.

Let me cite a few noteworthy instances of recent designs, and of research and development work, by officers of the Bureau of Steam Engineering.

Propellers.—The diagram, showing the estimated and actual performance of a battleship propeller, is largely typical in indicating the recent progress made by Rear Admiral Dyson in solving this complex problem.

Destroyers.—Immediately after the declaration of war by this country, our building facilities were concentrated on light craft—destroyers, submarines, patrol boats, and mine sweepers—to meet the menace of German submarines and mine layers. The figure shows diagrammatically the arrangement of turbines with reduction gears for the propelling machinery of the destroyers. This arrangement is unusual in the compact installation of engines of 27,500 horsepower on an average floor space of 26 x 56 feet.

Patrol Boats, "Eagles."—Because of their chief duty—hunting submarines—the machinery of the "Eagles," built by Henry Ford, was required to be as smooth running, vibrationless, and noiseless as possible, in order to secure the most favorable conditions for "listening" in submarine detection. This object was attained in the Bureau's design by omitting reciprocating parts almost wholly in the main and auxiliary machinery, nearly all motion being rotational. An officer who was present during the trial of the first of these boats, says that it gave him a peculiar sensation to be on a vessel speeding at 18 knots with so little sound and vibration.

RESEARCH AND DEVELOPMENT.

Some very striking advances of high military value in radio and submarine apparatus and in aircraft equipment are due directly to the research and development work conducted under direction of the Bureau of Steam Engineering during this war.

Submarine Detectors.—Since the war began eight devices for submarine detection have been employed by the Bureau and made in quantity, not only for our own Navy, but for the British Admiralty and the French Ministry of Marine.

Modified forms of these devices have also been designed for use in the protection of the Atlantic coast of the United States.

In the experimental investigation of this problem the scientists who worked on it under the direction of the Bureau of Steam Engineering realized from the beginning that the noises emitted by the submarine gave the most feasible means of locating it, since water is an excellent conductor of sound waves. Therefore, virtually all of the instruments thus developed and used to hunt German submarines are acoustical in nature. The successful results which have been obtained show the wisdom of research and invention along these lines.

The chief desideratum was, of course, to produce a device which would enable the pursuing vessel to determine the location of the submarine and then destroy it. It turned out, however, that the chief function of these devices was to render the submarine powerless, rather than to make its destruction possible. When the commanders of German submarines learned, through bitter experience, that, as long as their motors were running, the location of their boats could be ascertained by any nearby vessel equipped with detection apparatus, they ceased taking hazardous chances.

Instead of going where they pleased and making surprise attacks at will, they found that it meant almost certain destruction to keep underway in the vicinity of enemy vessels. The result was that German submarines spent a large share of their time resting on the bottom, where they were as incapable of doing harm as if they had never left their docks. It is apparent, therefore, that the number of submarines actually destroyed is not an accurate index of the efficacy of these submarine detection devices.

Briefly, the reason why a submarine must lie motionless to avoid detection when the enemy is near, is that the underwater boat when underway gives out a distinctive noise which can be recognized at once by the listener stationed at the detection device of the hunting vessel. This noise is similar to that

made by a phonograph when the record has been played and the needle is traveling through the inner grooves of the disc. It has been often noted also that a submarine emits a periodic "squeak." These and other distinguishing characteristics make it possible for the listener to detect a submarine, even in the presence of other vessels underway at the same time.

Obviously, since the matter is a military secret, the various forms of anti-submarine devices cannot be explained in detail. It may be said, however, that the first types of these apparatus were rigged outboard, and that, in the second stage of their evolution, the devices were towed astern or rigged through the hull. All of these methods, however, proved inadequate, since it was impossible for the hunting vessel to use the apparatus while underway. More recent experimentation has developed a device in which the "ears" are made an integral part of the hull of the vessel, and hence will serve while underway.

Considerable progress in the perfection of these devices was being made when hostilities ended. While the ideal detection instrument has not yet been evolved, it is wholly probable that further experiments, now under way, will result in the development of devices as far in advance of the present types as the latter exceed the first crude detection instruments.

Helium.—Helium is the mysterious gas which, camouflaged as "Argon," the Bureau of Steam Engineering and the Bureau of Mines of the Interior Department strained every nerve to produce in quantity during the last six months of the war.

The hydrogen employed for floating observation and dirigible balloons has one vital defect—its inflammability. When struck by an incendiary bullet the balloon instantly bursts into volcanic flame and falls in smoking fragments to the earth. Helium, on the contrary, is an inert gas like nitrogen. Its use in balloons means, therefore, that the fire risk is eliminated, and that a helium-filled airship could be brought down or seriously injured only by driving an airplane into it.

Helium is a relatively rare gas, and it is both difficult and

costly to separate it from the mixtures in which it occurs in natural deposits. Until the year 1918 there had never been more than five cubic feet of helium in any one container, and its cost had been from $1,500 to $6,000 per cubic foot. Our Government is now producing it at a rate of about $80 per thousand cubic feet, and shipments for use in kite balloons for the Army had been made just prior to the armistice.

Investigations, made prior to the war, showed that considerable quantities of helium existed in the natural gases of Kansas, Oklahoma and Texas. In the Province of Ontario, Canada, an appreciable amount of helium was also found in certain deposits of natural gas there. In any case, the most practicable means of separating the helium from the other components of the natural gas was the liquefaction of all of the gases except the helium, which could then be drawn off. This process involved the use of apparatus for obtaining temperatures as low as 317 degrees F. below zero.

While the United States remained neutral nothing was done with our deposits of natural gas containing helium. But as soon as we entered the war the British Government referred to our military and naval authorities various research problems whose solution would aid in ending the war. Among those problems was that of securing an adequate supply of helium for use in observation and dirigible balloons.

This question was considered by the Joint Army and Navy Aircraft Board, and the result of the deliberation was that the Bureau of Mines began the design of a plant, on original lines, in the endeavor to lessen the cost of separating helium, while the Bureau of Steam Engineering of the Navy Department insisted on modifying existing apparatus and, so far as possible, on securing the help of operating companies in obtaining the gas.

So the effort to produce "Argon," the camouflaged helium, was prosecuted vigorously. The Bureau of Steam Engineering enlisted two organizations in the matter, the only ones in

this country which had the necessary experience for such novel work. The Aircraft Board had made recommendations in October, 1917, and, in March, 1918, the first of these organizations had a plant in operation. This was followed in May by the completion of a unit by the second company. During the experimental period of the operation of these plants the Navy Department has been in close touch with all that was done and with all organizations which could aid in the work.

Experimental work has continued steadily with the earlier plants and that of the Bureau of Mines, which was completed in October, 1918. The production of helium in quantity was recommended by the Aircraft Board in August, 1918, and, when this recommendation was considered by the Secretaries of War and the Navy, it was decided that one agency should do the entire work, the Navy Department being the one designated. Since that decision was reached a production plant has been designed, under direction of the Bureau of Steam Engineering, by the organization which built the first of the two experimental plants noted above, and the construction of the buildings at Fort Worth, Texas, has been undertaken by the Bureau of Yards and Docks of the Navy Department. A pipe line to supply the plant with natural gas is also being laid. The first unit of this quantity-production plant should be in operation by next March. So America has solved the problem of helium for aircraft and observation balloons.

Submarine Radio Equipment.—A number of marked advances in Radio Telegraphy are due to the action of the Bureau of Steam Engineering. The work on the various systems described below has been done largely by outside parties in coöperation with the Bureau.

For example, take the radio equipment of submarines. The underwater radio equipment for these boats consists of a single-turn, insulated loop antenna. The loop is made up of two highly insulated wires, one of which is grounded to the

hull at the bow and leads aft to the conning tower and then down through a deck insulator to the radio room. The other wire is grounded to the stern of the submarine and leads forward to the conning tower and thence to the radio room.

This constitutes a closed loop having the hull for a return wire. This loop is connected to the standard radio equipment in the usual manner, except that a condenser is used in series with the loop when transmitting with the loop. It is possible to receive under water at depths reaching 20 feet by this apparatus. Sending under water has not so far been accomplished.

Subterranean Radio System.—The underground radio system experimented with, and from which the developments were made, was that of Mr. J. H. Rogers, of Hyattsville, Md., and consists of highly insulated wires laid in moist earth or water. These wires are laid at the four points of the compass from the receiving station, the outer ends of which are insulated. The ends of the wires in the receiving station lead up to a selector switch. This switch is connected to the receiving set, by the use of which various combinations of wires can be obtained. The underground wires are of different lengths for the different wave lengths. The standard receiving apparatus is connected in the usual manner. Efficient reception with this system is possible; it has directional qualities.

Radio Compass.—Let a wireless wave strike a rectangular loop, kept in a vertical plane and having a detector in its lower side. Also let the loop be capable of rotation on a vertical, central axis. Then, if the plane of the loop is perpendicular to that of the wave, the currents induced in it will neutralize each other, and the detector will register no sound. But, if the plane of the loop is at any other angle with that of the wave, the induced currents in its vertical sides will have the same direction, and a sound will be received through the detector.

This is the basic principle of the Radio Compass. In using it, the loop is swung, first, in one direction until the sound received from the detector is loudest; then in the other direction until the sound received there reaches a maximum; and, finally, the angle between these two maxima is bisected, and the plane of the loop if set at this mid position will be pointing directly at the station from which the wireless wave emanated.

This compass has been so perfected that it is possible to steer a ship into port in any weather by its use. Recently, one of our naval vessels got its position in foggy weather from the station at Cape May, 280 miles distant.

Radio for Aircraft.—Radio telegraphic apparatus for seaplanes has been developed which is efficient for ranges up to 200 miles, and for dirigibles up to 600 miles. Radio telephone sets, having ranges one-half those of the telegraphic sets, have also been developed. A trailing wire is used as an antenna.

Communication between planes in flight is possible up to ranges of 75 miles by wireless telegraph and 50 miles by telephone. The most satisfactory radio telephone for seaplanes is the product of the General Electric Company. Radio telephones have been installed on virtually all our naval vessels, the type used being principally that furnished by the Western Electric Company. The radio telephone has been especially valuable for manoeuvering hunting squadrons of submarine chasers and seaplanes.

A radio compass has also been developed by the Bureau engineers for the purpose of determining direction by radio signals received in an airplane. The instrument gives fairly accurate direction up to 500 miles when the plane is in flight and is exceedingly valuable for aircraft navigation.

REPAIR OF THE GERMAN MERCHANT SHIPS DAMAGED BY THE VANDALISM OF THEIR CREWS.

The almost incredibly rapid repair of the German merchant vessels lying in our ports and seized by our Government when

we entered the war makes one of the most striking and dramatic stories in the history of naval engineering. I am unable to give it in full, since the Secretary of the Navy will treat the matter at some length in his forthcoming Annual Report, and I may not anticipate his statement. I will, however, give some details—engineering mainly—which will probably not be embodied in that report.

The views which will be shown comprise: one showing the use of oxyacetylene welding in the repair of some of the large main cylinders of these ships, and others illustrating the repair by electric welding of broken parts of the machinery of the ex-German steamer *Santos.*

The *Santos,* I should state, is not one of the vessels seized by our Government. When war broke out, 46 German merchant ships were interned in Brazilian harbors. When Brazil entered the war its government seized these ships, but not before some damage had been done to their machinery by their crews. Later, Brazil turned over 29 of these vessels to the French Government. One of them, the *Santos,* could not be repaired with the facilities available at Rio Janeiro, since her machinery was virtually a heap of junk.

So the French naval attaché at Washington appealed to our Navy Department, and, at the suggestion of the Engineer-in-Chief, the broken parts of the *Santos* machinery were crated and shipped to the Philadelphia Navy Yard, where they were repaired by electric welding under the supervision of Captain Clarence A. Carr, Engineer Officer of that yard, through whose courtesy I am enabled to show the *Santos* views tonight. Their value lies in the fact that they form a complete series, illustrating fully the methods used by our Navy in repairing the ships we seized, except that, in most cases, our work was done on the broken cylinders while they were in place and the engines were not dismantled.

There were, in all, 103 of these German ships seized by our Government. Of this number about 50 were turned over to

the Navy by the Shipping Board for repairs. The major damage inflicted was the breaking of cast-iron parts of the main engines, chiefly the cylinders. But, in addition, some connecting rods, piston rods, and boiler stays were sawed through, boilers were burned out by dry firing, and there was much minor vandalism.

The chief problem, however, was the repair of the broken cylinders, some of which were more than nine feet in diameter. The Bureau estimated that with such facilities as were available for this heavy work it would take at least eighteen months to replace the damaged cylinders by new ones. Meanwhile there was bitter need of rapid transport for our troops to France. On the removal of that crisis there seemed to depend the salvation of Christendom.

At this juncture, Captain Earl P. Jessop, then Engineer Officer of the New York Navy Yard, recommended that repairs be made by welding, and in this recommendation he was heartily supported by Rear Admiral George E. Burd, Industrial Manager of that yard. The Engineer-in-Chief at once ordered Captain Oscar W. Koester, Assistant to the Bureau, to New York to investigate the situation fully. As the result of his report the Bureau immediately issued orders to make repairs, where possible, by electric or oxyacetylene welding, and where welding was impracticable owing to the location of the break, to resort to mechanical patching—that is, to "soft patches" secured by body-bound bolts.

From this stage onward the Engineer-in-Chief placed the repairs under the personal supervision of Captain Koester, who in his resourceful ability and driving power is unsurpassed. His work went on night and day, and in five months he traveled about 14,000 miles in railroad trains from one navy yard or private plant to another along our Atlantic coast. At the expiration of that period, his huge work was done and the ships were ready for service.

As soon as repairs were completed on each vessel the

strength of the new seams was tested under the full pressure of the original design, and none failed. She was then sent to sea on a minimum trial run of 48 hours under full power, the instructions to her officers being to break down the machinery by legitimate over-work, if they could. Every engine on every vessel withstood all tests, and the ships have been in continuous service since then.

Some few of these large cylinders—about ten, I believe—were repaired by oxyacetylene welding. This method was abandoned, however, not because of the quality of the weld, but because the method did not suit the conditions. With it the cylinder had to be dismantled and removed, and then, owing to the size of the flame, the huge mass of metal had to be pre-heated. These conditions did not exist with the electric arc, whose welding point is very small and can be localized, and whose temperature can be absolutely controlled.

As to the employment of the electric arc for this purpose, this method is too well known for description here. Its use, however, on such an extensive scale was unprecedented, and great credit is due the staff of the welding company who did so rapidly such perfect and, at the time, such vital work, and also to the officials of that company, who personally coöperated with the Navy in this great undertaking.

The rapidity of getting these ships ready for service was facilitated by one other action. On many of them, in addition to the damage done by vandalism, extensive repairs were necessary, owing to their deterioration from long idleness. As the necessary overhaul and all routine repairs could be made by a capable naval crew under able officers, the Engineer-in-Chief laid the matter before Admiral Benson, Chief of Operations, with the request that the ships be fully commissioned before repairs were begun. This was done, and thus while the welding operations were in progress the ships were prepared in all other respects for sea.

One other feature of this matter deserves comment, and

that is the military value of this work in the transport of troops. These fifty ships were in service for about a year before hostilities ended, and this is approximately the time saved in these repairs by using welding methods. Twenty of these vessels can carry about 70,000 troops in one trip, and ten round trips a year is a conservative estimate of their performance. Representative Young, of North Dakota, who crossed the Atlantic in the *Leviathan,* stated recently in the House that she had transported more than 99,000 troops up to August 20. So it may be justly claimed that the rapidity of these repairs had a marked effect on the early end of the fighting in France.

It is true that criticism was leveled at the Bureau for using these welding methods, and, too, by not a few engineers and shipbuilders. But the answer to all criticisms from every source is *that the ships are running,* without accident or delay.

THE ELECTRIC DRIVE FOR BATTLESHIPS.

In those stormy days toward the end of the Civil War, Isherwood, that master engineer, designed for the United States Navy the 17-knot cruisers *Wampanoag* and her sisters, which, as their trials proved, were incomparably the fastest ships, naval or merchant, at that time in the world. The jealousy and ignorance of those days of semi-sail power brought down on him a storm of criticism, and ultimately the great Engineer-in-Chief was forced to see his peerless ships rot at their moorings.

In the years since then there has been no fit comparison with this shameful episode in our naval history, except, in less degree, the similar storm which burst on the head of the present Engineer-in-Chief, when, with high courage and far vision, he urged the introduction of the electric drive for the propelling machinery of battleships and battle cruisers.

In one vital element, however, there was a radical difference between these two historic controversies. Isherwood was

faced by a preponderating opposition from the officers of the old sailor Line of the Navy, and, on the contrary, the adverse criticism as to the electric drive had no naval origin. By far the great majority of the Navy—its officers and officials—stood to a man behind the Engineer-in-Chief in his steadfast and virile insistence on this innovation.

The crux of the opposition lay apparently in commercial reasons. It is true that some eminent engineers in civil life, who had no financial interests in the matter, were wholly sincere in voicing their disapproval of what, with their inadequate knowledge of naval conditions, seemed to them a hazardous venture. But, as to others, whether the basic reason for the war waged on the Bureau of Steam Engineering in opposition to the electric drive was patriotic or pecuniary may well be left to those who made the attack. I may say, however, that the question of royalties was involved in the difference between the use of the turbine in driving the propeller shaft directly or through interposed gearing, and its use indirectly as a primary source of power for generating an electric current to drive that shaft through an electric motor.

The conspicuous feature of the controversy thus forced on the Navy Department was the vociferous propaganda conducted by the opponents of the electric drive, in telegrams and letters to, and interviews with, the Secretary of the Navy; in a letter to the President of the United States; in letters to the chairmen of the Naval Committees of the Senate and the House of Representatives; and in a number of other ways, including a letter to the Secretary from a distinguished professor at a great University, recommending—at the request, he said, of somebody else—a proposition made previously that a Board of impartial experts be appointed by the Secretary of the Navy to consider and advise on the whole matter—that is, to tell the Bureau of Steam Engineering just what it should do in its own business.

And, too, this propaganda had its laughable side. For ex-

ample, one prominent electrical expert wrote vigorously against the use of electricity in this way, and then, apparently under business pressure, wrote again, and just as vigorously, approving its use—to the huge amusement of naval officials. And, further, the editor of a technical journal made himself absurd by presuming to tell naval officers and the Navy Department the kind of ships it should build, and by attempting to arrogate to himself a knowledge of naval matters superior to that of the General Board of the Navy, which had examined and approved the design he criticized.

Throughout this controversy the Engineer-in-Chief held unwaveringly to his decision. No arguments had power to turn him, for he had what his critics lacked, detailed knowledge both of naval requirements and of the performance of electricity as a driving power in naval vessels. About seven years before, the naval collier *Jupiter* had been thus equipped by his predecessor in office, Rear Admiral Cone. As soon as she was completed, Admiral Griffin, then Engineer-in-Chief, sent the Assistant to the Bureau—Commander, now Rear Admiral S. S. Robison—to the Mare Island Navy Yard, with orders to try the *Jupiter's* machinery at sea in every conceivable way. The report of this officer was exhaustive, and therefore Admiral Griffin knew in every detail just how the electric drive would act in all conditions. So from an engineering viewpoint, the contest between the Bureau and its critics was an unequal one.

However, that controversy matters little now, except, in its finish, as a warning to possible imitators of a future day; for the contract trials recently of the *New Mexico*—the first of our battleships to be fitted with the electric drive—have confirmed fully the judgment of the Engineer-in-Chief. She was tested in many more ways than her contract called for, and in all she has shown herself not only wholly efficient, but—in her system of propulsion, her maneuvering power, and her underwater protection, the leading battleship of her day in the navies of the world.

Now, let us turn to the technical side of this question. First, I need hardly say here that the steam turbine has replaced the reciprocating engine for battleship propulsion. And I need scarcely add that the combination of a turbine—whose maximum economy is shown at 2,000 revolutions or more—and a propeller—which, in a battleship, should not exceed 180 to 200 revolutions—presents a rotational problem of some complexity.

Evidently an intermediate reducing mechanism is required. There are now a number of these reduction gears—mechanical, hydraulic and electrical. The view shows the arrangement used in one of our older battleships. She is a twin-screw vessel, the two propeller shafts being shown at the left side of the diagram. The high-pressure and low-pressure turbines for each screw are in series and are mounted on separate shafts, each shaft carrying a pinion which engages in single reducing gear of large diameter keyed to the propeller shaft. Each low-pressure turbine shaft carries also an astern or backing turbine.

The principal objection to this arrangement is the necessity for, not only a backing turbine on each screw shaft, but one which gives only about 40 per cent of full power, since questions of weight and of space forbid the use of a backing turbine of the same size as the ahead turbine, and hence the result is a short one operating at low economy. A minor, but important, disadvantage is the very considerable width in a transverse or athwartship direction of the machinery space.

Now consider the electric drive, as shown diagrammatically by the view which follows. It will be seen that while the turbine is still the primary source of power, it operates an electric generator instead of a screw propeller, and, as the view of a four-screw ship shows, the electric power thus developed is transmitted to motors on the propeller shafts in much the same way as electric power is transmitted on shore. In other words, there is a central power plant—composed of

boilers, turbines and electric generators—and the power is taken from this plant to the motors on the screw shafts.

This installation possesses an advantage over the direct or geared-turbine drive, in that a backing turbine is not required. The reversing is done through the motors with the same ease and certainty as in any other electric motor installation, and with the further advantage that full power can be utilized in backing.

With a suitable construction of the motor, the turbines can be driven at maximum economy for both the full speed and the cruising speed of the ship; for other speeds, the turbine and alternator must be slowed down in the usual manner. This arrangement therefore gives a two-speed, reversible motor at maximum economy in both speeds and with full power in backing.

The elimination of the backing turbine is a matter of prime importance, since most of the accidents to turbine-driven vessels of the Navy are due to it. For example, the dreadnaught *Arizona,* in her first trip from New York, had been making a very fine record in straight-away steaming, but when she began maneuvering one of her turbines came to grief, and she had to limp back to the New York Navy Yard, with a big job of re-blading to be done.

The electric drive has numerous other advantages, which I cannot now recount. In sum, its use gives:

Greater flexibility and control in the use of power.

Improved economy in every-day service and good economy at high speeds.

Less liability to serious derangement of machinery.

Less likelihood of the speed of the ship being seriously affected in the event of injury to the turbines.

And, these two paramount advantages:

Military superiority from greater maneuvering power;

Military superiority due to increased under-water protection.

In the noble future which the destiny of nations foreshadows for this Republic two things seem clear to our vision now: first, that the United States is to have a huge merchant marine, perhaps the largest in the world; second, that for guarding that commercial fleet and to do our share in keeping world-peace we are to have also a Navy at least equal to that of any other nation. So any advance which adds to the efficiency of our fleet is of fundamental value to us.

This the electric drive does. Its advent has, in fact, revolutionized the propelling systems of the battleships of all navies. Secretary Daniels has written, in courtly phrase, of "the vision and the wisdom of Rear Admiral Griffin" in foreseeing all this; but it should not be forgotten that the Secretary of the Navy, to his everlasting honor, stood loyally by the Engineer-in-Chief throughout this bitter controversy. And, too, we owe tribute to a distinguished engineer, Mr. W. L. R. Emmett, an Annapolis graduate, for his steadfast advocacy of the electric drive, and, as well, to the great electrical organization on whose staff he is, and whose enterprise and business sagacity made this advance possible.

CONCLUSION.

I have endeavored this evening to give in swift survey some account of the achievements of the Bureau of Steam Engineering during this war. The scope and magnitude of its work have been so great, however, that my resumé has been necessarily both general and incomplete. And yet, one fact should be clear: that an immense amount of work has been done, very rapidly and with full efficiency.

As to that work I may say that, while I have no authority to speak for the Engineer-in-Chief, I am sure that I voice his views when I state—and with pleasure before this Society—that, although all of this work has been done under direction of the Bureau, it could not have been accomplished without the hearty coöperation and assistance of those ship and engine-

building establishments which delivered battleships, destroyers, submarines and submarine chasers to the Navy during the war.

Still less would this work have been possible without the whole-hearted support of the large manufacturers of steel, and of copper and brass, who were the first to place their products at the disposal of the Government. To the great electric companies, to a large manufacturer of boilers and another of pumps, and to the smaller manufacturers, whose name is legion and whose output is to be found on every ship, great credit is due for their aid in making successful so stupendous an undertaking.

And, last, but far from least, our Navy Yards and the many private plants whose facilities have been utilized in repairing the machinery of our ships, and in thus maintaining the efficiency not only of the Fleet but of the vessels engaged in the transport of troops and munitions, deserve the highest praise for the energy and efficiency which has characterized their work during this war.

I have added pleasure in this tribute here, since so many of the men interested in this vast aggregation of engineering and shipbuilding enterprises are members of this great national Society.

DYNAMICS OF GEAR DRIVE.

BY N. W. AKIMOFF.

In a recent remark in the "International Marine Engineering" I gave a short outline of the problem of the rational design of the gear drive.

There is absolutely no escape from the fact that, so far, the design of this drive has been based only on the *Static* considerations; the *Dynamics* has always been left severely alone, assuming therefore—

1. That the shaft is *infinitely short*, and just as *infinitely rigid.*

2. That neither the gear nor the propeller weigh anything at all, or that their radii of gyration are next to zero.

3. That, instead of the propeller we have something like a generator, under an absolutely constant load.

I will ask the reader, how reasonable these assumptions really look. The whole thing boils down to this: *do we want to know what we are doing or don't we?* Here are the facts:

1. The shaft is generally both long and comparatively slender.

2. Both the gear and the propeller are exceedingly heavy, and their moments of inertia are tremendous.

3. The resistance offered by the propeller is not constant but periodic, each blade contributing one maximum and one minimum during each revolution; so that, for instance, for a three-bladed propeller we have three impulses, of some kind, per revolution. Just what these impulses are is another matter, not in the least interesting from the immediate standpoint of the present discussion. Prof. G. K. Calhoun, U. S. N., thinks that the predominant cause of these variations lies in the proximity of the rigid boundary (the hull proper); while, on the other hand, I personally feel that the variation is

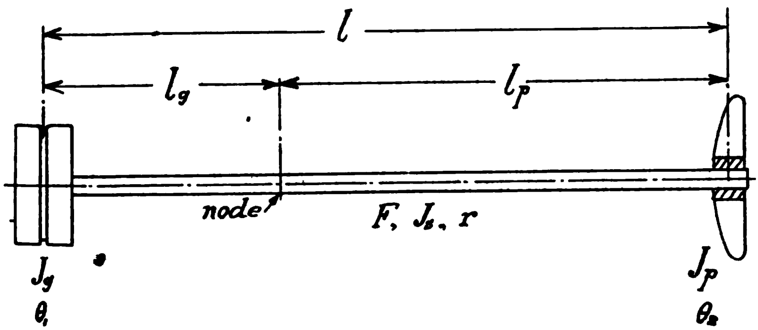
largely due to the variable submergence of the revolving blade, wrapped up in its vortex, all of which will be submitted later, in a separate communication. From Admiral Dyson's remark on page 107 of his book on Propellers, it appears that tip clearance has a great deal to do with vibrations in general.

At any rate these three (or whatever the number might be) impulses per revolution, are somewhat in the nature of external periodic force (we say *force*, in the generalized sense, of course), the magnitude of which we neither do nor care to know, but whose frequency, at least, is something tangible.

They are probably not very great in themselves, but if their frequency is the same as (or near) that of one of the natural modes of the angular, torsional, vibration of the elastic dynamical system, *gear-plus-shaft-plus-propeller*, then synchronism will take place and there will be trouble.

So, if we really mean to know what we are doing we must either make sure that such synchronism will not take place; or, if it cannot be avoided, for some reasons, that we have at our disposal certain means whereby to paralyze its effect.

The most important part of the problem, therefore, is to know at least the fundamental frequency (the gravest mode) of the system shown in the sketch.



The fact that the shaft does not float in the air but is mounted in bearings, does not matter very much; friction, unless abnormal, does not materially change the rate of free vibrations.

That such torsional vibrations actually take place has been known for the past eighteen years. Dr. Hort's beautiful book, *Technische Schwingungslehre*, gives a fairly complete bibliography of this subject; in more recent years the matter of torsional vibrations has been looked into, from experimental standpoint by our Navy Department; and the article on the subject, by Naval Constructor Wm. McEntee, published in this Journal, is no doubt well familiar to its readers.

However, most of these papers and articles deal with engine drive, assuming the resistance of the propeller to be constant and not periodic, and dealing with considerable amplitudes.

We shall therefore derive a complete solution of the problem in a manner that could be adapted to be of immediate benefit to the designer of a gear drive:

Problem. The length of the shaft is l, the polar moment of inertia of its cross-section is J_s; the area of cross-section is F; the modulus of elasticity (shear) is G; and the mass of one linear foot of shaft is γ, the radius of gyration of the cross-section being r. The weights of the gear and the propeller are W_g and W_p; their moments of inertia are J_g and J_p. Everything in feet and pounds. *Find the frequency of the fundamental mode of the torsional vibration of such a system.*

Solution. Taking, at random, any cross-section of the shaft, whose distance from one of the ends is, say, x, and denoting its angular deflection from the normal position by θ, we shall readily see that the unit-deflection (angular) will be $\frac{d\theta}{dx}$, so that the torque, or moment, due to elastic forces, will be

$$M = GJ_s\frac{d\theta}{dx}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

In an adjoining section, dx apart from the one we considered above, the moment of elastic forces will differ somewhat from the above; it will be

$$M + dM = GJ_s\left(\frac{d\theta}{dx} + \frac{d^2\theta}{dx^2}dx\right) \quad . \quad . \quad . \quad . \quad (2)$$

(by Taylor's series).

Hence an immediate solution of the problem of the motion (torsional) of the slice dx itself: indeed, the moving or external force (in our case *moment*) will then be

$$dM = GJ_s \frac{d^2\theta}{dx^2} dx, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (3)$$

while the *mass* (rotational) will be simply the moment of inertia of the slice dx, that is

$$\gamma r^2 dx;$$

the *acceleration* of this element (of course *angular*) will be $\frac{d^2\theta}{dt^2}$ and since the principle *force=mass × acceleration* applies to rotational motion, if only instead of *force* we take the corresponding *moment;* instead of *mass* the *moment of inertia*, and, instead of linear *angular* acceleration, we have

$$GJ_s \frac{d^2\theta}{dx^2} = \gamma r^2 \frac{d^2\theta}{dt^2}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (4)$$

Dividing by J_s, which of course is $= Fr^2$, we have

$$\frac{GF}{\gamma} \frac{d^2\theta}{dx^2} = \frac{d^2\theta}{dt^2};$$

or, representing, for convenience, $\frac{GF}{\gamma}$ by a^2, we finally have

$$\frac{d^2\theta}{dt^2} = a^2 \frac{d^2\theta}{dx^2}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (5)$$

This is a very well known partial differential equation, generally referred to as *equation of vibrating strings;* although it is really the starting point of all problems on elastic vibrations in general.

Even before we have found a solution of this equation we can readily see that the lowest tone, or fundamental mode, of such a vibration will comprise *one node*, or position of rest, somewhere along the shaft, so that both ends will move in opposite directions, reaching their extreme positions at the same time; also passing through their central position simultaneously.

For an unloaded shaft of uniform cross-section the node will naturally be in the middle; for a shaft, having heavy masses on its ends, nothing can be stated, beforehand, regard-

ing the position of the node, except that there will be one, somewhere.

The particular solution of the equation (5) is sought in the form

$$\theta = A. \sin pt. \sin mx. \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

Here the constant A has something to do with the actual amplitude, and from our standpoint is only of secondary interest. But the constants p and m are of immediate importance: p is the frequency-constant and m apparently has something to do with the form of the curve (sine curve in this case) which will be the diagram of the angular twist (*not* the elastic curve, as would be the case in a problem on transverse vibrations).

In order to see if there is any connection between p and m, such as would enable them both to satisfy our tentative solution (6), let us substitute the latter into (5), upon proper differentiation as to x and t. We have, upon obvious reduction

$$p^2 = a^2 m^2, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

which establishes the relation between p and m.

We shall now look into the *end conditions:* let us assume that the location of the node is established by l_g and l_p, both lengths so far unknown.

Consequently the torsional moment on the left end will be found by integrating (3) between zero and l_g; this will give

$$M_g = GJ_s \left(\frac{d\theta_1}{dx}\right)_{x=l_g}$$

This very moment is operating on the gear, whose moment of inertia is J_g; applying therefore the same principle (and in the same sense) as above, we have *force=mass×acceleration*

$$GJ_s \left(\frac{d\theta_1}{dx}\right)_{x=l_g} = -J_g \frac{d^2\theta_1}{dt^2}, \quad . \quad . \quad . \quad . \quad (8a)$$

and, in the same manner, for the other end

$$GJ_s \left(\frac{d\theta_2}{dx}\right)_{x=l_p} = -J_p \frac{d^2\theta_2}{dt^2}. \quad . \quad . \quad . \quad . \quad (8b)$$

The sign — means that the moment (in our case the elastic resistance) tends to bring the body toward zero position (of

rest), as is the case in all elastic vibrations. Utilizing the form (6) adopted above, we have

$$GJ_s \sin. pt. \cos. ml_g = J_g p^2 \sin. pt. \sin. ml_g, \quad . \quad . \quad (9)$$
$$GJ_s \sin. pt. \cos. ml_p = J_p p^2 \sin. pt. \sin. ml_p,$$

or, simplifying and remembering that

$$p^2 = a^2 m^2, \text{ also that } a^2 = \frac{GF}{\gamma},$$

we have these two equations in their final form

$$m \tan (m.l_g) = \frac{J_1}{J_g} \quad . \quad . \quad . \quad . \quad . \quad (10)$$
$$m \tan (m.l_p) = \frac{J_1}{J_p}$$

where J_1 is, of course, the moment of inertia of one linear foot of the shaft about its axis.

To these we must add the obvious relation

$$l_g + l_p = l, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (11)$$

and these three equations will determine the three unknowns of our problem, namely l_g, l_p and m.

These are found by the method of successive approximations. First, we assume a certain value of l_g and find the corresponding value of m from the first equation (10); this value of m will now be substituted into the second equation (10) and the result solved for l_p. If the value of l_p thus found does not check up with the condition (11), another trial should be resorted to.

According to the remark made by Prof. Timoshenko (of St. Petersburg) in his paper on shaft vibrations, we should always begin by the following approximate value of l_g

$$l_g = l \frac{J_p + \frac{lJ_1}{3} \cdot \frac{J_p}{J_g + J_p}}{J_g + J_p + \frac{lJ_1}{3}}. \quad . \quad . \quad . \quad . \quad . \quad (12)$$

Having found a satisfactory value of m we must utilize it for finding the desired frequency per minute, N.

The following reasoning will be found quite obvious since

$$p^2 = a^2m^2 = \frac{GF}{\gamma}m^2, \text{ we have}$$

$$p = \sqrt{\frac{GF}{\gamma}}\, m = \frac{2\pi}{60} N,$$

so that

$$N = \frac{30}{\pi}\sqrt{\frac{G}{\gamma}}\, m \sqrt{F} = 103{,}000\, m \sqrt{F} \quad . \quad . \quad (13)$$

The reader, if interested, will check the following numerical example:

$J_1 = 23.2$; $J_g = 12{,}500$; $J_p \doteq 3{,}550$; $F = .8$ sq. ft. $l = 122$ ft.

Here m comes out to be $= .0077$.

The position of the node is given by

$$l_g = 31 \text{ ft}; \; l_p = 91 \text{ ft};$$

and, finally, the number of free torsional vibrations per minute will be

$$N = 103{,}000 \times .0077\sqrt{.8} = \text{about } 710.$$

This means that, for a three-bladed propeller, the speed of 236 R.P.M. will lead to synchronism—and therefore to trouble.

In solving equations (10) and (11) a 20 inch slide rule can be used to advantage; also a table of natural values of circular functions similar to that given on page 20–22 of Hoüel's *Recueil de formules.* (Paris, 1901).

ELECTRIC WELDING PLACED ON A SCIENTIFIC BASIS.

By Captain of Engineers J. H. Chalker, U. S. C. G., Member.

In order to insure a perfect weld the elements which constitute the electrode and the object to be welded must be analogous, and this can only be determined by research and experiment. For instance, take two pieces of steel from the same plate and weld them; then place the welded piece in a testing machine and pull until ruptured; if the steel is found to have, for instance, a tensile strength of 60,000 pounds and an elongation of, say 25 per cent in eight inches, this steel upon analysis would probably show about .162 to .182 carbon and about .34 manganese, but to be certain, the steel constituting the test piece as well as the electrode should be analyzed.

It is a well known fact that with ordinary open-hearth steel ship and boiler plates the amount of carbon and manganese are the deciding factors governing the tensile strength and elongation of the material, although both tensile strength and elongation can be raised or lowered by cold or hot rolling.

Now continue the experiments with testing machine and analyses of steel of various tensile strengths and elongations, electrodes of known elements and an electric current of known voltage, and ampèrage.

The carbon and manganese contained in electrode should bear a certain relation, depending upon the strength of the weld, to the carbon and manganese in the metal to be welded; that is, the per cent of carbon and manganese in the electrode should be high enough to allow a certain portion of both to be burned out in the process of making the weld and still contain the necessary amount to insure a strong weld. For instance,

it might be necessary to use an electrode containing .29 per cent of carbon and .60 per cent, or upward, of manganese, while the weld when completed might contain but .17 per cent carbon and .32 per cent manganese. If all the carbon and manganese contained in the electrode were burned out the tensile strength of the weld would probably not be greater than that of wrought-iron while it should approach the tensile strength of the material welded.

After a series of tests and analyses together with currents of various voltages and ampères a point is arrived at where a weld ceases to be, so to speak, an experiment in any sense of the word, the only undetermined factor being the thickness or bulk of plate to be welded. The welder is no longer obliged, except in isolated cases, to have the material which is to be welded analyzed. It is only necessary to refer to the table which has been compiled to know which electrode, voltage and number of ampères to use for steel of known chemical constituents. In other words, knowing the constituents of the metal to be welded one would simply use an electrode and current, determined by these experiments, best adapted to assure a perfect weld.

Referring to the electric welding of certain parts of the steam machinery of German ships under the supervision of the Navy Department, the work was performed under circumstances similar to those outlined in this paper; that is, a constant potential circuit was used, the same being not over 40 volts open circuit. The current was regulated for a predetermined value and automatically controlled so as not to vary more than 5 per cent either way.

Of course it is realized that the skill of the welder will always be a dominating factor in good welding, but this might be termed a commercialism and a case of "you pay your money and you take your choice," for the best welders should command the highest compensation.

It is only necessary to know the heat or melt number, which

is stamped on each ship and boiler plate, to obtain from the manufacturer both the physical and chemical tests, and as a matter of fact this data is usually required by ship builders with the shipment of plates from the mill.

The same experiments should be made with cast iron and other metals to be welded, and thus place electric welding on a secure and scientific basis and eliminate all guessing.

These experiments could be conducted by the Bureau of Standards, with various welding machines or perhaps the better procedure would be for the manufacturer of each type of welding machine to conduct his own experiments together with his own analyses, and furnish the necessary data with each machine placed on the market.

The compilation of data would not be either a long or an expensive task, for the range of physical and chemical requirements for steel plates, shapes and castings are not great, say between 60,000 and 70,000 pounds tensile strength, and as before stated the amount of carbon and manganese is in direct proportion to the tensile strength and elongation, the carbon being the governing factor. The higher the tensile strength of the steel the less will be the elongation.

Phosphorous and sulphur should be as low as possible, but in open-hearth steel made under the basic process neither the phosphorous or sulphur is usually high.

OPTICAL METHOD OF SHAFT ALIGNMENT.

BEING A REPORT OF REALIGNMENT OF INBOARD PROPELLER SHAFTS U. S. S. "MISSISSIPPI," NORFOLK NAVY YARD, JUNE, 1918.

By Captain William Norris, U. S. Navy, Member.

The U. S. S. *Mississippi* had reported excessive vibrations, under way, in the vicinity of the starboard inboard propeller, with leakage around strut palms and hull plate seams. As the spring bearings of No. 2 shaft had occasionally given trouble, it was believed desirable to check alignment of this shaft at the first opportunity.

The *Mississippi* was docked, June, 1918, the starboard inboard propeller and stern-tube shafts removed, tested in the shop, and found true within .010″. A line, the usual wire, was centered in the inboard and outboard stern-tube bearings and carried through the strut. The strut appeared to be so far out of line that it was decided to recheck, using the optical method recently developed at this Yard (described later). Measurements taken checked very closely with the ordinary line, the measurements showing the center of the bore of the struct 7/16″ low and 3/16″ outboard, in the docked condition, with respect to the centers of the stern-tube bearings.

The port inboard propeller and stern-tube shafts were then removed, and alignment checked as in the case of the starboard inboard shafting, the center of the strut in this case being down 9/32″ and outboard 1/8″.

It was thought that this condition might be caused by docking, as the ship had been on the blocks for several days without the overhang of the stern being supported from the dock floor. At this time, other urgent docking work required that the *Mississippi* be undocked, and, during the undocking, the

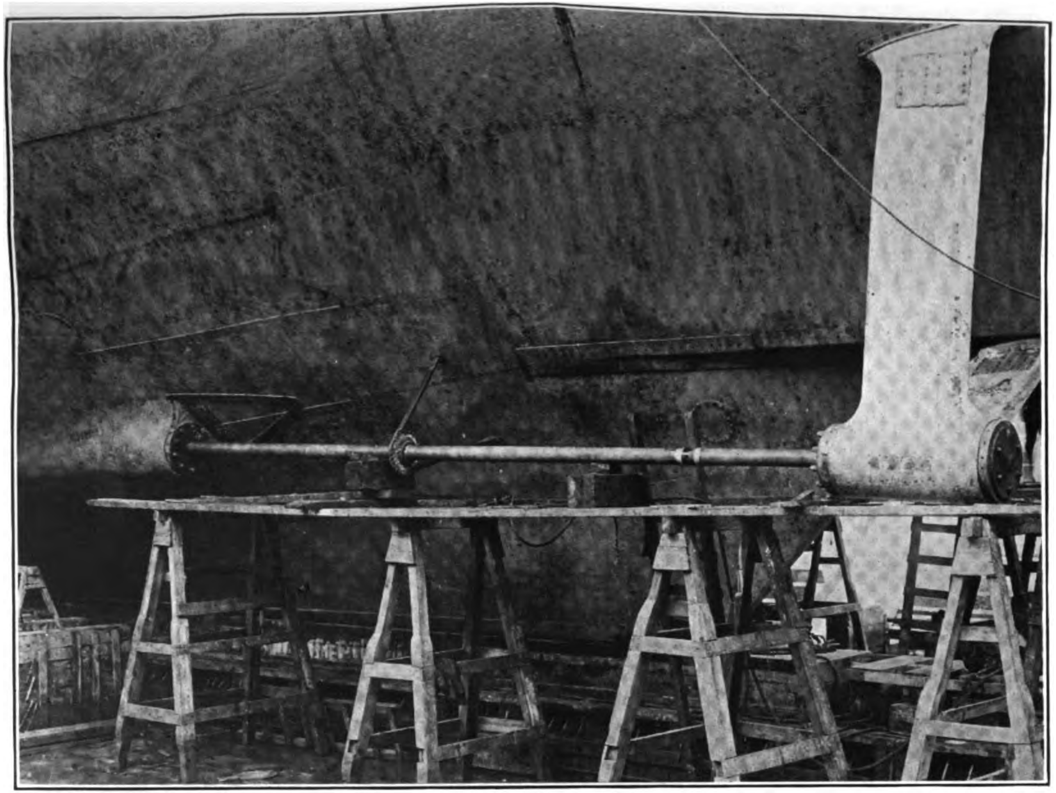


PLATE I.

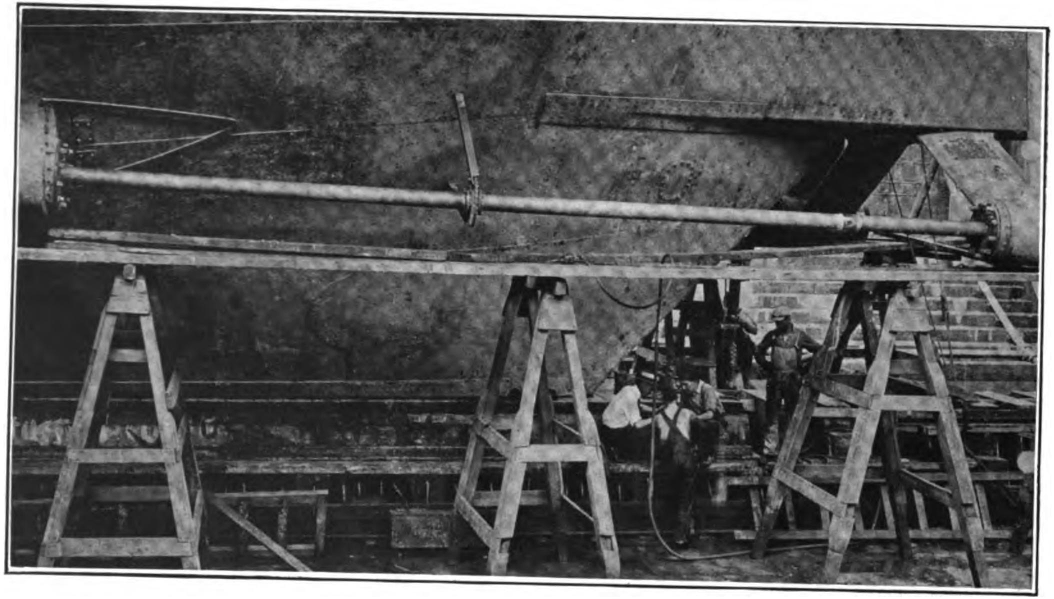
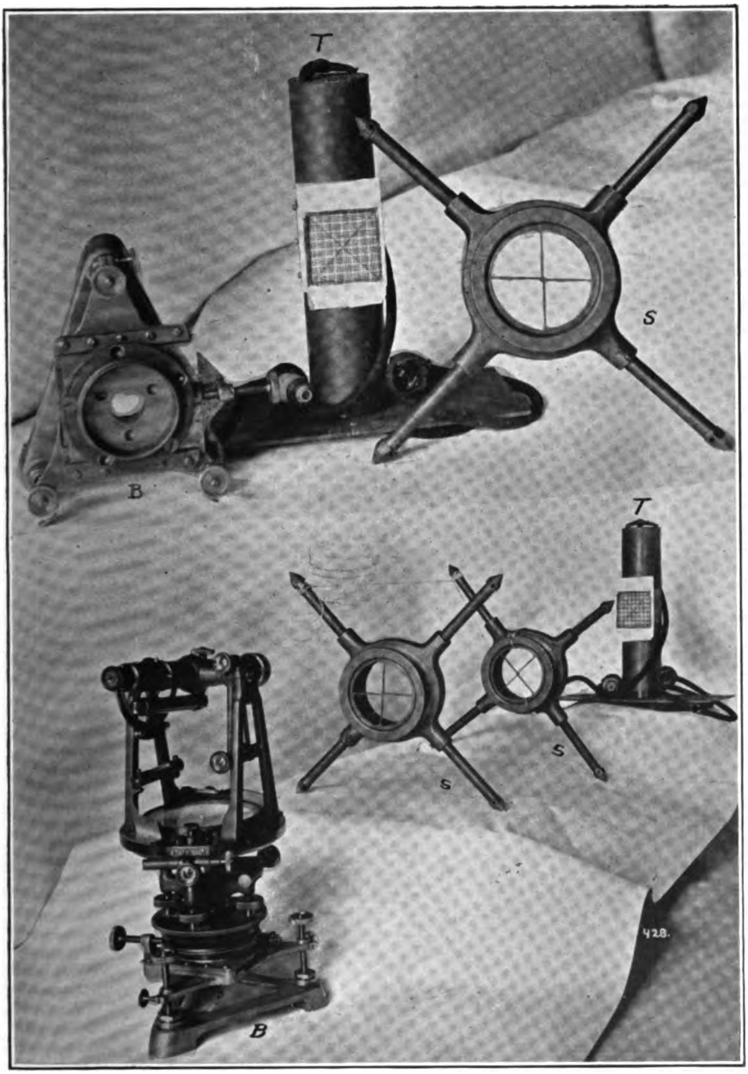


PLATE 2.



PLATES 3 AND 4.

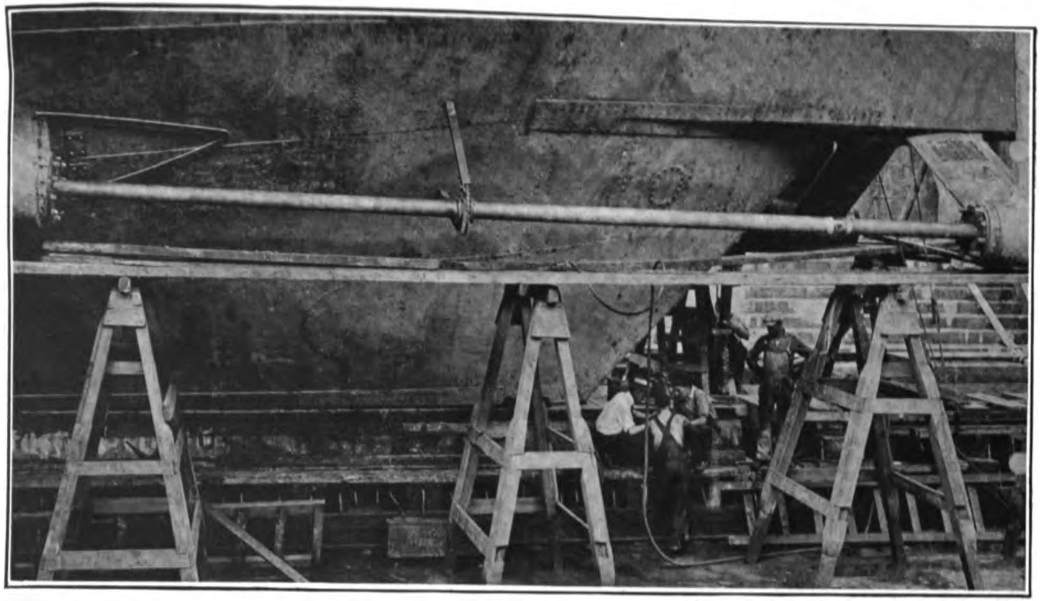
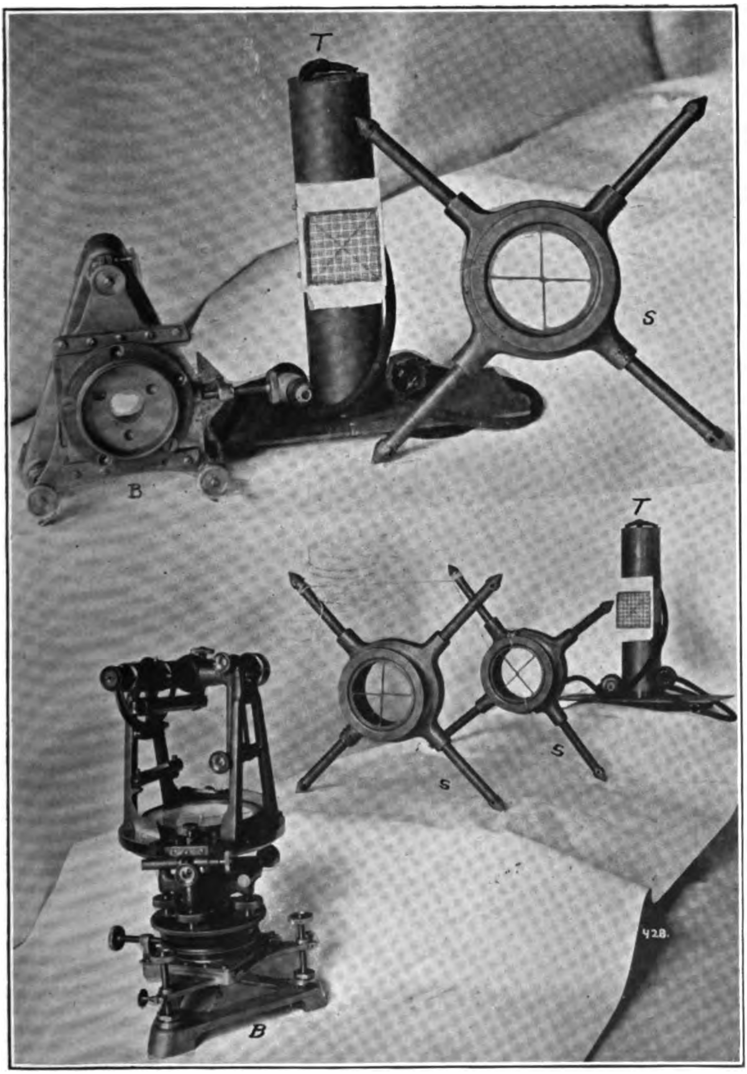
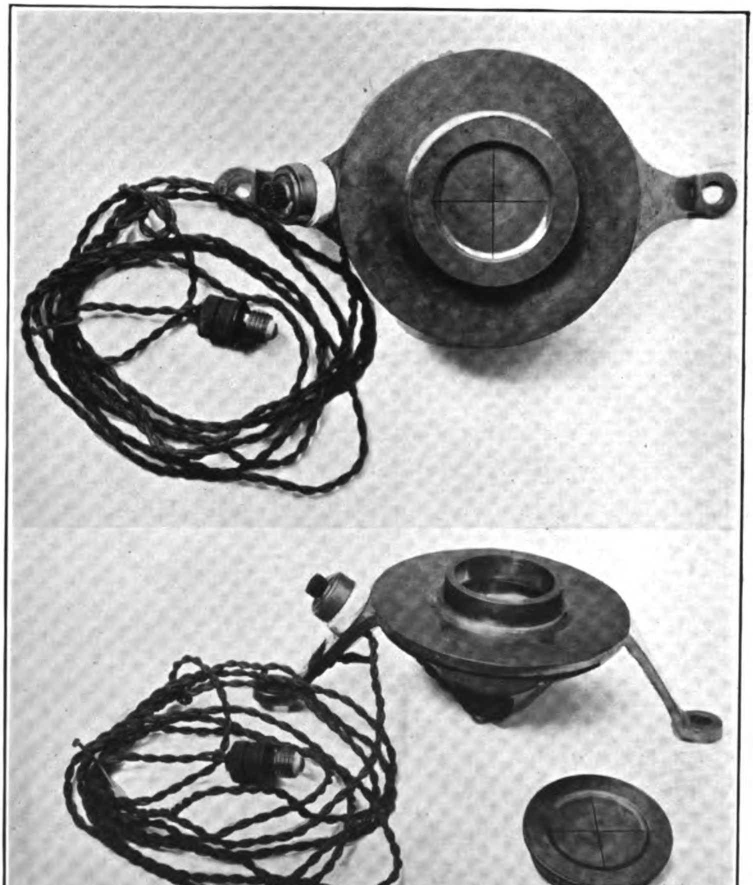


PLATE 2.



Plates 3 and 4.



PLATES 5 AND 6.

opportunity was taken to determine whether or not the stern would rise or fall relative to the structure as a whole when the vessel was water borne. The result of this investigation indicated an actual sagging of the stern when the vessel floated. Measurements of this deflection were taken with a surveyor's level mounted on deck close to the after turret, readings being taken before the dock was flooded and with the vessel afloat.

Two facts were thus determined: first, as docked, the struts for Nos. 2 and 3 shafts were considerably out of line with reference to the stern-tube bearings; secondly, there were indications of sagging of the stern relative to the hull as it had rested on the blocks when the vessel was floated. It was thus necessary to determine the alignment of the strut with respect to the stern-tube bearings *with the vessel afloat;* and the method of determining this is described as follows:

In general, the scheme was to project a beam of light from a target located in the strut through the stern-tube to a surveyor's transit located in the shaft alley, a 6″ watertight tube (see Plates Nos. 1 and 2) being fitted between the forward face of the strut hub and the outboard stern-tube bearing, the after face of the strut hub being blanked off. Spiders carrying perforated centers were located at the exact center of bore of both inboard and outboard stern-tube bearings. Then by locating a transit so that the cross-hairs were in line with the centers of the stern-tube bearings, a line of sight was established which, when projected on to the target in the strut, showed directly the deflection of the center of the strut with reference to the stern-tube bearings.

Plates 1 and 2 show the watertight tube from strut to stern tube and the method of securing, also the flexible flanged joint to take up any changes in the hull structure, allow for expansion and any deflection that might be due to buoyancy of the tube. As a precaution in case the tube should carry away, the inboard end of the stern tube was also blanked off, a hole being

left to lead the electric-light wires through to the target and a three-inch hole through which to sight the transit, the latter hole being tapped and a pipe plug, and also a wooden plug, kept close at hand for use in emergency. After the tube had been rigged and before the dock was flooded, air pressure was put on the stern tube, connecting tube and struts to test for leaks and several holes were found in the hub of the strut, which were plugged. The target (Piece T, Plates 3 and 4) was secured in the strut before the connecting tube was secured in place. The target consists of a 3-inch diameter brass tube 12 inches long, mounted on a piece of brass plate, and carrying on the inside of it two electric lights (carbon). A 2½-inch square section is cut out of this tube, and the target proper is fastened over the opening. The target consists of a framework of brass with 1/32 inch holes drilled on 1/16-inch centers around the sides of the frame and waxed linen thread laced through these holes as shown on Plate 3. The particular system of coördinates used is to facilitate the reading of the exact point where the cross-hairs of the transit strike the target. The center of the target, the intersection of the two diagonal threads, was located at the center of the bore of the strut hub. The electric light leads ran through the tube and stern tube. The centering spiders (Pieces S, Plates 3 and 4) were located in the inboard and outboard stern-tube bearings with their centers coincident with the center of bore of the bearings. These two spiders formed locating points for establishing the line of sight.

After locating the spiders in the stern tube and the target in the strut hub, the expansion tube and blanks were bolted in place, the air test for tightness made, and the ship floated.

The transit was adjusted approximately at any convenient time after the stern-tube spiders were installed and secured, and checked after the vessel was afloat. A special base plate (see Piece B, Plates 3 and 4) was found very convenient for quickly locating the transit.

With the above equipment in place and the light turned on the target, it was a simple matter to set the transit in line with the centers of the spiders in the stern tube by alternately focusing the transit on each center. This established the line of sight, and it was then only necessary to focus the transit on the target. The point where the cross-hairs of the transit intersected the target indicated the true center line of the stern tube extended. The amount this point differed from the intersection of the diagonal threads on the target was the amount the center of the strut hub was out of line with the center line of the stern tube, and this amount could be read directly from the target. In applying the necessary correction of misalignment, its direction will be determined by the type of transit used, whether inverting or erecting.

Plate 4 shows the sequence of setting up the various parts, and also the necessity of cutting away the caps of the spiders to allow focusing through the first spider to the second spider and from there to the target.

The usual precautions for handling a precision instrument must be observed in setting and focusing the transit, but with reasonable care and an accurate instrument, the maximum error should not exceed 1/32 of an inch at 125 feet, with a transit magnifying 24 diameters.

Measurements taken on the *Mississippi, the vessel afloat,* with the apparatus described above, showed that the center of the port inboard propeller strut hub was 13/64″ outboard and 3/8″ lower than the center line through the stern-tube bearings; and that the center of the inboard starboard propeller hub was 3/16″ outboard and 3/8″ low.

Readings of this misalignment were taken both with the ship on the blocks before flooding and immediately after flooding, and it was observed that the sagging of the stern upon floatation of the ship was not more than 1/16 of an inch. Plate 7 shows blue prints of the targets in these instances, the dotted

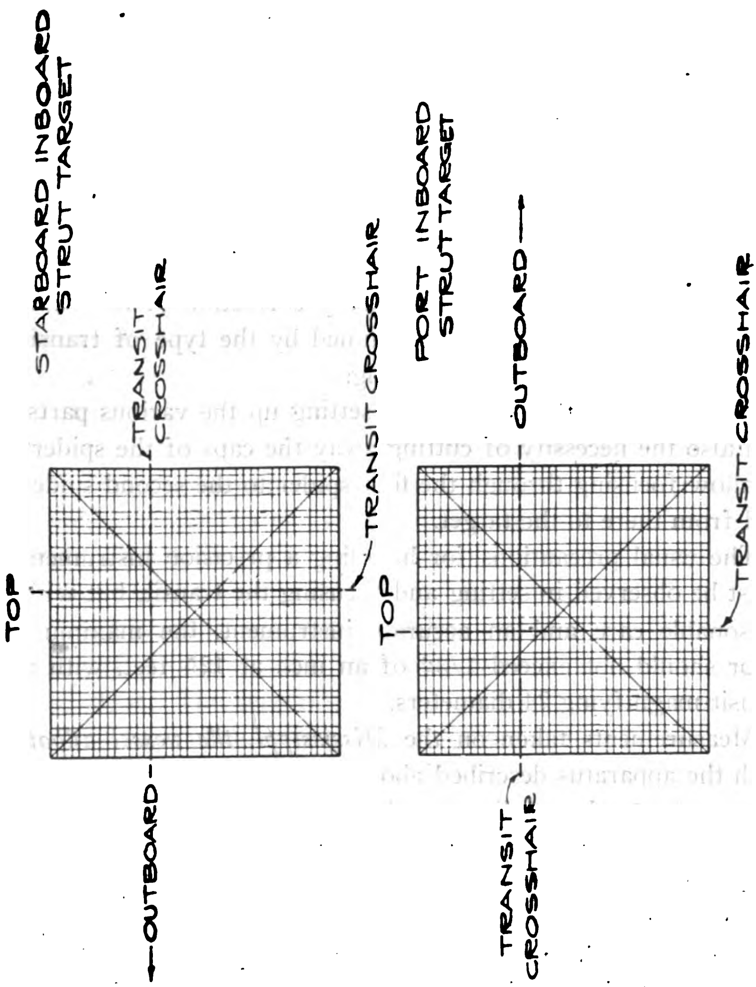

STARBOARD INBOARD
STRUT TARGET
TRANSIT
CROSSHAIR
TRANSIT CROSSHAIR
TOP
OUTBOARD
PORT INBOARD
STRUT TARGET
OUTBOARD
TRANSIT CROSSHAIR
TOP
TRANSIT
CROSSHAIR

PLATE 7.

line indicating the location of the transit cross-hairs on the target with the ship afloat.

The errors noted were corrected by reboring struts and fitting new strut-bearing bushings.

In conclusion, it is stated that this method of optical alignment has been successfully used at this Yard for establishing the shaft lines on vessels under construction. If the coördinates of any two points along the shaft line are given—for instance, a point on a bulkhead and the location of the center of the stern-tube bearings—the location of all the other bearings along the shaft line can be determined, it being only necessary to set up the transit in line with the two given points, and, without changing the setting of the transit, focus it on any other point in line and set the centers of the spiders coincident with the cross-hairs of the transit. The center of the spiders then form a locating center from which to lay off points preliminary to boring.

Instead of using the target which is described above, when establishing a shaft center line, a special target (see Plate 5) or light-cross is used. As will be seen from Plates 5 and 6, this target is a bracket-like device on the back of which is mounted a parabolic reflector carrying a low candlepower electric light and on the front a cap. Through this cap are cut two 1/32″ slots forming a cross through which the light from the reflector is projected onto the lenses of the transit. This target is mounted on a bulkhead with the center of the light-cross as one established point in the center line of the shaft. The cross-hairs of the transit may be very accurately located in the light-cross and the reflected light from the target (with the cap removed) allows of the work being carried on at night when the hull is of a uniform temperature and interference by other workmen on the ship is eliminated.

The shaft lines of Destroyer No. 70, *Craven,* were thus located, the time required for both shafts being about three hours. The alignment was later found to be very good.

The details and general arrangement were perfected by Mr. L. C. Campbell, Junior Inspector of Naval Construction, to whose interest and initiative the successful development of this method of shaft alignment is largely due.

AN IMPROVED METHOD OF OPERATING EVAPORATORS.

By M. C. Stuart,* Associate.

Marine Engineers will agree that the evaporating plant has the unenviable reputation of requiring more attention and causing more trouble than any other of the ship's auxiliaries. The determination of the proper coil pressure, water level and other conditions for the production of a desired capacity forms a source of continual speculation. Priming, reduction in capacity and the rapid accumulation of heavy masses of scale are the accompaniments of the most careful operation. Recent improvements in design of shell and coil leave untouched the many problems and difficulties of operation.

A new method of operating evaporators, developed at the U. S. Naval Engineering Experiment Station, Annapolis, Md., shows such remarkable and positive results in solving the questions and lessening the difficulties of operation as to warrant its careful consideration, and, it is believed, its adoption, by Engineer officers who are interested in maintaining the ship's fresh-water supply with the least possible effort.

The essential feature of the method is the production of fresh water at a constant rate and at any desired capacity within the limit of capacity of the evaporator. This result is obtained by the application of a steam orifice to the coils, which, by reason of the law of constant flow of steam, as expressed by Napier's formula, effects a constant production of vapor without the need of additional regulating appliances. In addition to the constant capacity feature, the use of the method reduces to a minimum the liability of priming, con-

---

*Mechanical Engineer, U. S. Naval Engineering Experiment Station, Annapolis, Md.

siderably lessens the difficulty with scale, and stabilizes and simplifies the entire operation.

Before describing the method it will be desirable to review the process of evaporation, and discuss at some length the usual method of operation, in order that the principle of the new method and the manner in which it remedies some of the difficulties and defects of the usual method may be clearly understood.

The production of fresh water from sea water is accomplished by the vaporization of sea water by means of the condensation of steam in coils which are surrounded by the sea water or brine contained in a shell. The condensed steam is drained from the coils and the vapor is condensed in a separate distiller condenser, usually called the distiller. In a multiple-effect plant the vapor produced is carried along to serve as steam in the coils of a second evaporator or effect. As evaporation continues, the water level of the brine in the shell must be maintained by the continuous introduction of sea water as feed. This increases the amount of salt in the brine and the accumulated salt must be removed, either by the continuous discharge of a small amount of brine (continuous blow-down), or by the periodic discharge of the entire volume of brine when the salinity reaches a certain limiting value (intermittent blow-down).*

HEAT TRANSFER COEFFICIENT.

An unavoidable result of the process of evaporation is the gradual accumulation of scale on the brine side of the coils. This scale greatly affects the ability of the coil to transmit heat, and is one of the most troublesome features of evaporator operation. The ability of the coil to transmit heat is measured by the heat transfer coefficient, and inasmuch as the entire process of evaporation is so closely bound up in the

*For a complete discussion of blow-down, see "Salt Water Evaporators," by W. L. De Baufre, A. S. N. E. JOURNAL, Nov., 1915, page 946.

transmission of heat through the coil the heat transfer coefficient may be considered at some length with profit.

By definition, the heat transfer coefficient is equal to the B.t.u.'s transferred per hour per square foot of surface per degree temperature difference. This may be expressed by the formula,

$$K = \frac{W\,(H_s - q_d)}{S \times T.\,D.},$$

in which,

$K =$ Heat transfer coefficient.
$W =$ Steam condensed in coils, pounds per hour.
$H_s =$ Heat in steam entering coils, B.t.u. per pound.
$q_d =$ Heat in condensate from coils, B.t.u. per pound.
$S =$ Evaporating surface, square feet, measured on the outside of the coils.
$T.\ D. = T_s - T_v$, in which, T_s is the temperature corresponding to the pressure of the steam in the coils, and T_v is the temperature of the vapor corresponding to the vapor pressure in the shell.

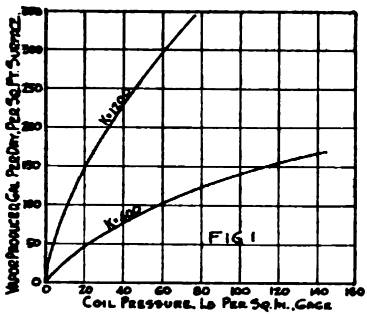
The numerical value of the coefficient for any given set of operating conditions may be computed from the equation. Physically, the value of the coefficient depends upon a number of factors, which, in the probable order of their importance, are condition and amount of scale accumulated, salinity of brine, water level, design of coils, vapor pressure and coil pressure. In a given evaporator, values of K ranging from 1,400 to 100 are possible. One thousand two hundred is an average value when the coils are clean and with sea water in the shell, and 200 is a probable average value when the scale has accumulated on the coils to a considerable extent.

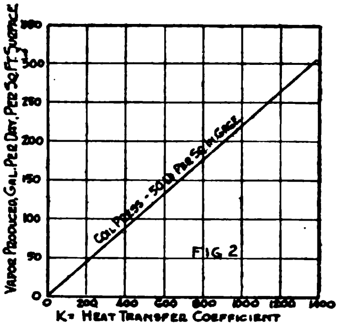
In the practical operation of an evaporator by the usual method a most important problem is the determination of the proper coil pressure for the production of a desired capacity. The temperature difference required for the production of any

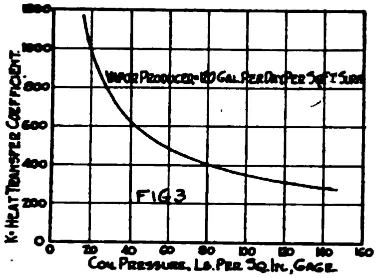
capacity may be determined at any time from the heat transfer coefficient equation if the value of K be known. Having the temperature difference, the coil pressure corresponding to the temperature difference is easily determined. Figure 1 shows the relation between coil pressure and capacity for a single-effect plant, with atmospheric shell pressure and values of $K = 1{,}200$ and 400. The curve for $K = 1{,}200$ will apply approximately when the coils are clean and scale free. As scale forms on the coils the value of K decreases, and at a constant coil pressure the capacity will decrease as shown by the curve of figure 2. This immediately suggests that in order to maintain the capacity as the value of K decreases it will be necessary to increase the coil pressure. The relation between heat-transfer coefficient and coil pressure necessary to produce a given constant capacity is shown by the curve of figure 3. The procedure of continually changing the coil pressure would be very troublesome and is seldom used. What is usually done is this: A coil pressure is selected which will give the capacity desired when the coils are partially covered with scale. This pressure is then maintained on the coils at all times by means of a reducing valve in the steam line to the coils.*

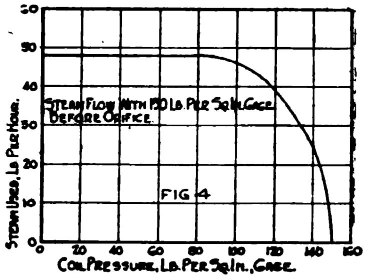
When the coils are clean the capacity produced is considerably above the normal amount, and in most evaporators the starting must be made very cautiously on account of the danger of priming, which is a natural result of high capacity. In order to minimize the danger of priming it is a usual practice to carry either a lower pressure or a lower brine level at starting. On account of the trouble of changing the reducing-

*The question of proper coil pressure is treated by Comdr. H. C. Dinger in an article, "Notes on Evaporating Plants," A. S. N. E. JOURNAL, August, 1914, page 849, as follows: "When the coils are clean a much lower pressure will suffice than when heavy with scale. As coils become dirty increased pressure will have to be used to secure the same capacity. It is, however, deemed inadvisable to be continuously changing the reducing-valve pressure. It would appear better to keep a certain pressure, which, in the double-effect plants of the Navy, should be about 60 pounds, and run with this all the time. When the capacity is materially reduced by salt deposit the evaporator should be scaled."









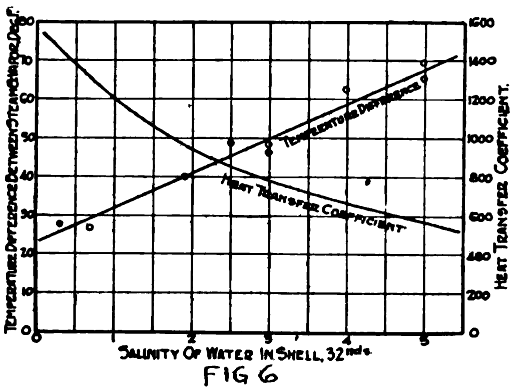


FIG. 1.—COIL PRESSURE AND CAPACITY.
FIG. 2.—HEAT TRANSFER COEFFICIENT AND CAPACITY.
FIG. 3.—COIL PRESSURE AND HEAT TRANSFER COEFFICIENT.
FIG. 4.—STEAM FLOW THROUGH ORIFICE.
FIG. 6.—EFFECT OF SALINITY OF BRINE.

valve setting the water level is usually chosen as the variable, and the operation is started with a low water level.

It is considered by some necessary to carry the water level low at all times in order that some exposed surface will serve to superheat the vapor, and therefore produce salt-free vapor. Others consider that submerged coils are an essential to the best results. It has been demonstrated without a doubt that it is highly desirable, for a number of sound reasons, to carry the water level high enough to fully submerge the coils, if this is at all possible. However, with the usual method of operation, it is almost always impossible to start operation with the coils submerged and the operating pressure in the coils, on account of the priming which is caused by the high capacity produced with clean coils. But, as the operation proceeds at constant coil pressure, the capacity falls off due to the decreasing heat-transfer coefficient caused by the accumulation of scale.

An evaporating plant should be considered as an apparatus for the production of fresh water at any desired rate, rather than an apparatus upon which certain condition of coil pressure, water level and salinity must be maintained while allowing the capacity to be governed by these conditions, as well as being at the mercy of the continually decreasing heat-transfer coefficient. The introduction of the steam orifice for the regulation of the steam flow to the coils satisfies exactly the requirements necessary for the desired constant production of vapor.

An orifice is installed in the steam line to the coils in place of the usual reducing valve. Full main-line pressure is carried on the inlet side of the orifice, and the orifice discharges directly to the coils. The steam which flows through the orifice is condensed in the coils, and only condensed steam is allowed to discharge from the coils, the use of a drain pot or trap insuring this. The pressure in the coils automatically adjusts itself to a value which will produce a temperature

difference sufficient to condense all the steam which enters the coils. When the coils are clean it is evident that a much smaller temperature difference is necessary to transmit the constant amount of heat involved in the condensation of the constant amount of steam than when the coils are covered with scale. With a constant line pressure the quantity of steam which flows through the orifice will be constant as long as the pressure (absolute) on the discharge side of the orifice is equal to or lower than the critical pressure, which is 0.58 of the line pressure (absolute). The amount of flow is determined very closely by Napier's formula,

$$W=\frac{P\,A}{70},$$

in which,

W = steam flow, pounds per second.
A = area of orifice, square inches.
P = absolute pressure on entering side of orifice, pounds per square inch.

Figure 4 shows the relation between steam flow and final or coil pressure for an initial line pressure of 150 pounds gage. For final pressures up to 81 pounds gage the flow is constant; for final pressures higher than this the flow falls off at first slowly, and then more rapidly, till the final pressure equals the initial pressure, when the flow is, of course, zero. For final pressures higher than the critical pressure the flow is determined by a thermodynamic formula which need not be considered here, for we are concerned only with coil pressures lower than the critical pressure.

The temperature and pressure in the coils may be determined by again having recourse to the formula for heat transfer coefficient.

$$K=\frac{W(H_s-q_d)}{S\times T.\,D.}.$$

In the orifice method of operation the steam flow, W, is constant. The total heat in the steam "H_s" is constant. The

heat of the liquid of the condensed steam "q_d" increases only slightly with the coil pressure. Therefore, (H_s-q_d), which is the heat removed by condensation in the coils, B.t.u. per pound of steam, is also practically constant. The surface, "S," is constant, and K, the coefficient of heat transmission, is variable, depending upon the condition of the coils as regards scale and certain other conditions of operation The temperature difference may be solved for as follows:

$$T.D. = \frac{W(H-q)}{K \times S}.$$

Since K is the only variable on the right side of this equation, it is evident that for every value of K there is a corresponding value of temperature difference required. The shell pressure fixes the shell temperature, and, with the temperature difference known, the coil temperature and coil pressure are easily found. The coil pressure will adjust itself to whatever value is required to produce a temperature difference which will be necessary to transmit the heat through the coils.

The operation of an evaporator with the orifice would therefore start with a high value of K and a very low coil pressure. As the operation continues and the value of K decreases the coil pressure will increase, and at the same time the steam flow, and therefore the capacity, remains constant.

TEST OF DOUBLE EFFECT PLANT WITH ORIFICE.

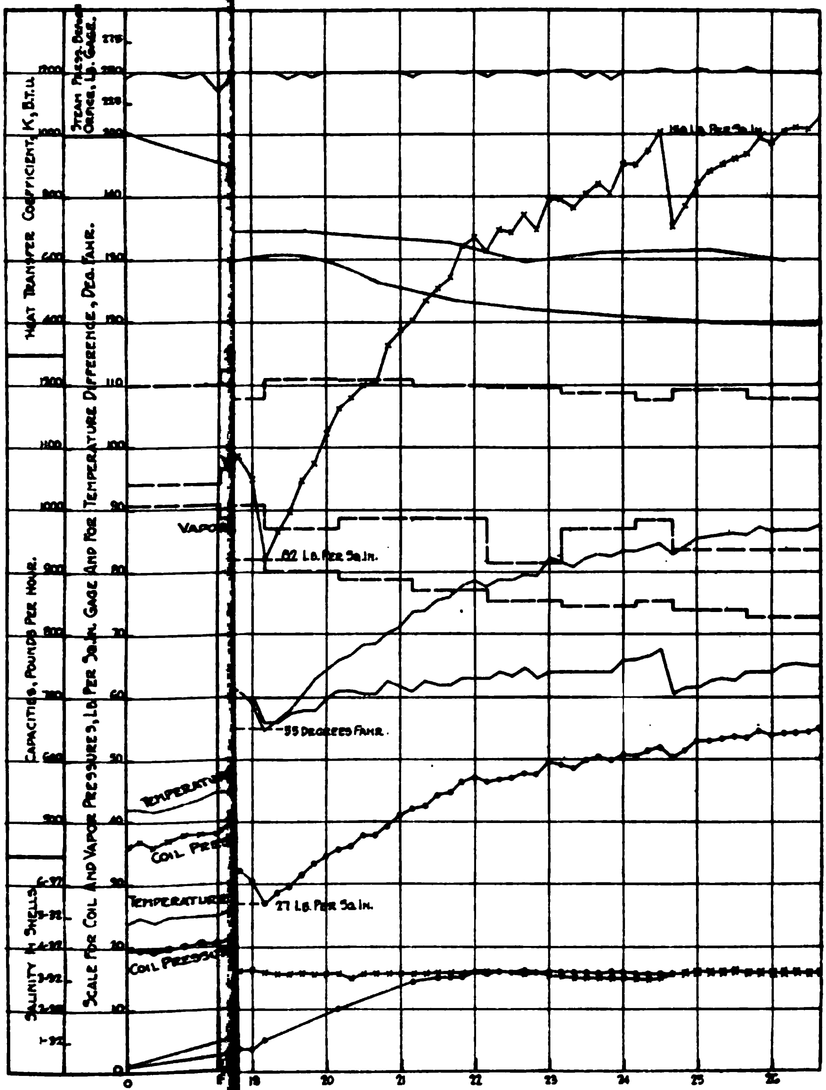
Having shown the theoretical action of the evaporator equipped with a steam orifice, the results of tests of the method will be given before discussing details of the application of the method. A preliminary trial was made upon a single-effect evaporator equipped with an orifice, and the action was exactly as was expected. At the start, when the coils were clean, the coil pressure was low, and as the operation continued the coil pressure slowly increased and the capacity remained constant. It was considered very desirable, however, to conduct the principal test upon a double-effect plant in view of

the wide use of the double-effect plant in the Naval Service and its added efficiency. A double-effect plant was therefore arranged for test. Each shell contained about 24 square feet of evaporating surface in the form of coils connected between manifolds as shown in the photographs, figures 7 and 9. A steam pressure of 250 pounds was available in the main line. The second-effect vapor discharged to a distiller condenser through a vapor feed heater, and atmospheric pressure was carried on the second effect shell. The vapor feed heater heated the feed for both effects to about 195 degrees, and the feed for the first effect was further heated by a coil drain heater. The capacity at which it was desired to operate was 120 gallons of water per day per square foot of total surface, which is double the present rating for Navy evaporators. From the expected efficiency of the double-effect plant this would require about 1,200 pounds of steam per hour to be supplied to the first effect coils. From Napier's formula, $W = \frac{PA}{70}$, for 250 pounds steam pressure in the line, the orifice required was computed to be 0.335 inch diameter. A brass plate 1/8 inch thick, with a hole in the center 0.335 inch in diameter, was placed between two flanges in the steam line to the coils of the first-effect evaporator.

The full line pressure of 250 pounds was carried on the line leading to the orifice. Atmospheric pressure was carried on the second-effect shell. The water in each shell was carried at such a height as to submerge the coils. This level was determined by trial, as will be explained later. A water seal was kept on the coil drain by means of traps part of the time, and drain pots part of the time. This insured that all the steam entering the coils was condensed therein and only condensate allowed to discharge from the coils.

The log plot of the results of the first run is given in the insert, figure 5. On this are plotted the coil pressures, temperature differences, capacities, heat transfer coefficients and

salinities for each of the two effects. The salinity on the first effect was allowed to increase to 3/32nds and was maintained at that value throughout the test. The salinity of the second effect was varied, in order to observe the effect of a variable salinity on the action of the evaporator. The second-effect salinity was allowed to increase to 5/32nds, was then slowly decreased to that of Severn River water, which is less than 1/32nd, was increased to 5/32nds again, reduced to below 1/32nd, and increased to 3/32nds. Throughout the series of 26 hours the steam flow to the plant was constant except for a slight falling off when the line pressure fell below 250 pounds at the 6th hour. The pressure in the second-effect coil started at 20 pounds gage and increased to 52 pounds gage at the 6th hour, when the salinity was 5/32nds. Upon the reduction of the salinity the pressure fell and followed the salinity till at the end of the 26th hour the pressure was 55 pounds when the salinity was 3/32nds. The first-effect coil pressure started at 36 pounds gage and rose and fell, of necessity following the second-effect coil pressure (which was the first-effect shell pressure). A first-effect coil pressure of 100 pounds was reached at the 7th hour. It then decreased slowly to 47 pounds, increased to 150 pounds, decreased to 82 pounds, and reached 150 pounds at the 26th hour, with a salinity of 3/32nds on both effects. The temperature difference for the second effect varied, following the second-effect salinity, but the temperature difference for the first effect increased uniformly throughout the series from 24 degrees at the start to 65 degrees at the finish. *Throughout all this period the capacity of both effects was practically constant,* such variations as took place being due to variations in heat losses of the blow-down and the coil drain. (It should be noted that the run was not made non-stop. The plant was shut down each night and the salt water allowed to remain in the shell. In the morning the operation was resumed, and in each case the conditions obtained on starting in the morning corresponded


Heat Transfer Coefficient, K, B.T.U.
Capacities, Pounds Per Hour.
Salinity in Shells
Scale for Coil and Vapor Pressures, Lb. Per Sq. In. Gage and for Temperature Difference, Deg. Fahr.
Vapor
.82 Lb. Per Sq. In.
55 Degrees Fahr.
27 Lb. Per Sq. In.
Temperature
Coil Press
Temperature
Coil Pressure

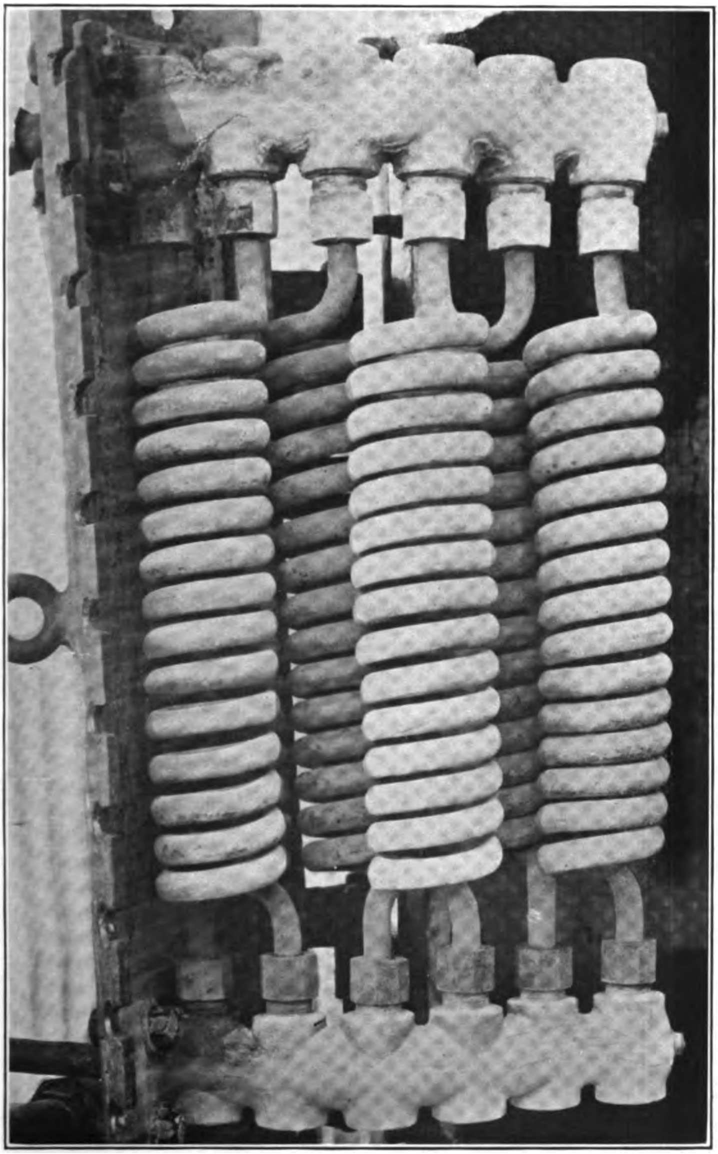
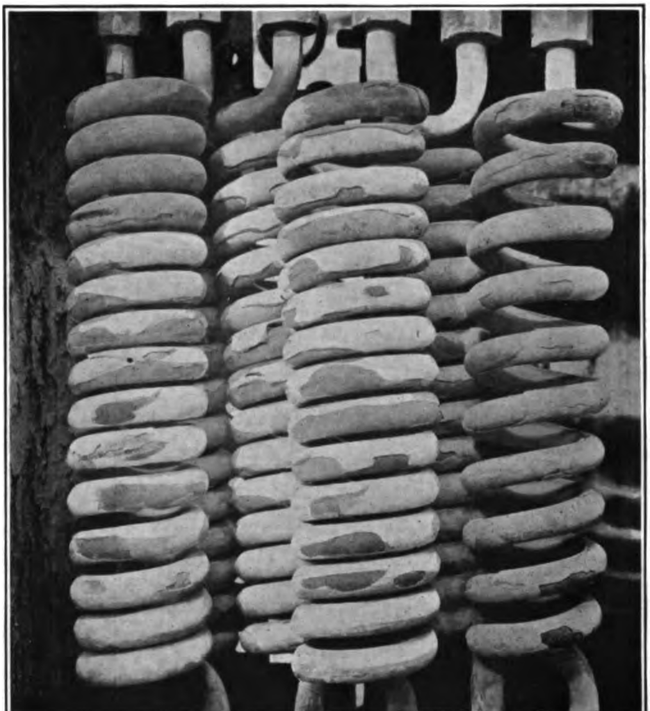
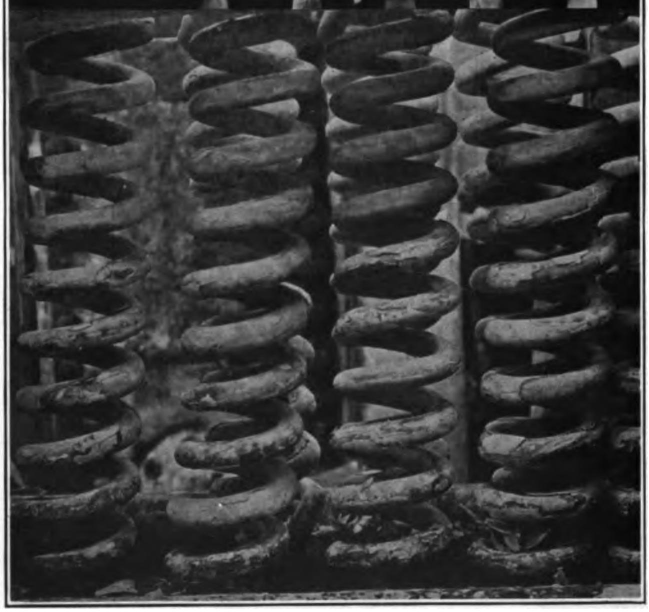


FIG. 7.—COILS, SHOWING SCALE AT END OF RUN WITH ORIFICE.





UPPER—FIG. 8.—COILS AFTER CRACKING SCALE, SECOND EFFECT.
LOWER—FIG. 9.—COILS AFTER CRACKING SCALE, FIRST EFFECT.

Fig. 10.—Scale in Bottom of Shell After Cracking Scale.

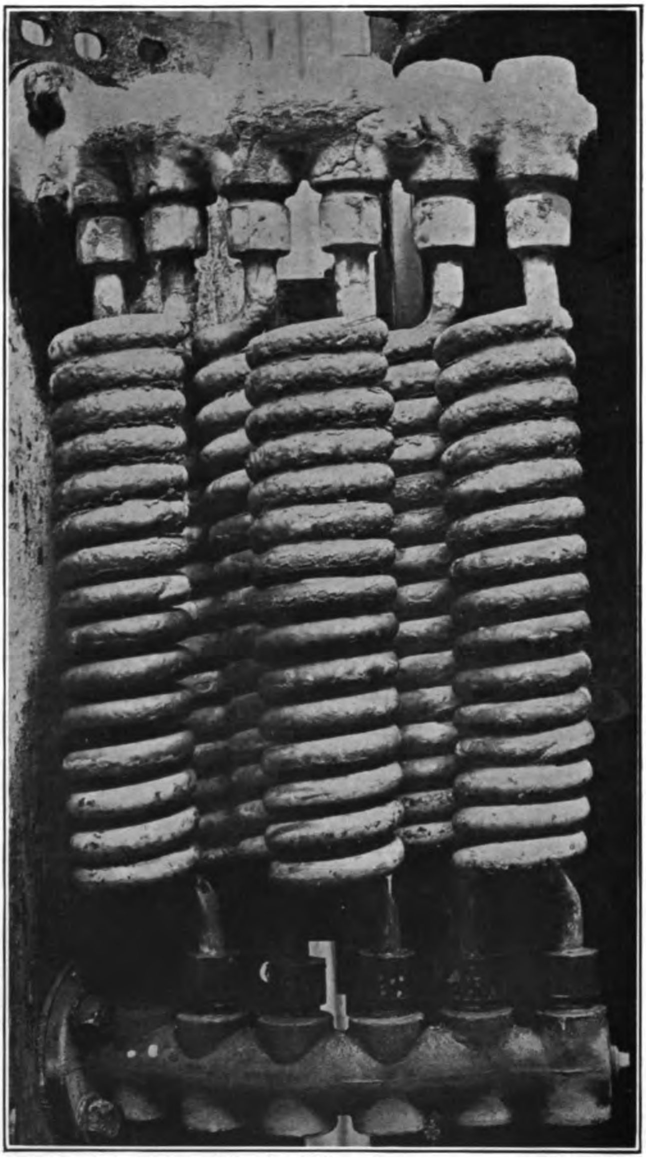


FIG. 11.—COILS, SHOWING SCALE PRODUCED BY LOW WATER LEVEL.

to the conditions which existed at the close of the previous day's run.) The average capacities obtained were as follows:

Steam used	1,196.6	lbs. per hr.
Vapor produced, 1st effect.....	984.6	lbs. per hr.
Vapor produced, 2nd effect....	896.4	lbs. per hr.
Total vapor	1,881.0	lbs. per hr.
Total vapor	543.	gals. per day
Total vapor	112.	gals. per day per sq. ft. of surface
Total vapor	1.575	lbs. per lb. of steam

The capacity produced was almost double the present rating, and there was no variation in capacity with variations of salinity from less than 1/32nd to 5/32nds and variations of first-effect coil pressure from 36 pounds to 150 pounds per square inch. No regulation of coil pressures was required. The vapor produced was of a purity of less than 1/2 grain chlorine per gallon at all times.

When the coil pressures reached the maximum values, operation was ceased for the removal of scale. Before removing scale, however, the second effect evaporator was opened and the photograph shown in figure 7 was taken, which shows a thin, even coating of scale on the coils. The evaporator was immediately closed and the scale was removed by a scale-cracking process which consisted of surrounding the coils with cold water and admitting 40 or 50 pounds steam pressure suddenly into the coils, and at the same time keeping the coils cleared of condensate and air by watching the drain pot. Upon opening the evaporator it was found that a large part of the scale had been cracked from the coils and was seen lying in the bottom of the shell. Fragments gathered from the shell are shown in figure 10.

The result of the removal of scale is an increase in the heat transfer coefficient, and upon resuming operation after removing scale the coil pressures are low and the cycle may be repeated. The question naturally arises as to the time required

for the pressures to increase to such values as to necessitate the removal of scale, and the influence of rate of evaporation and other operating factors upon this rate of increase of pressure. The temperature difference in the first effect increased at an average rate of 1½ degrees per hour. The coil pressure and heat-transfer coefficient may be determined from the increase in temperature difference.

The effect of salinity on temperature difference and heat transfer, as determined from the second-effect data, is given on figure 6. The temperature difference is increased about 9 degrees per 32nd increase in salinity and the heat-transfer coefficient is correspondingly reduced. At 1/32nd salinity the value of K is 1,200, at 3/32nd it is 780 and at 5/32nd it is reduced to 570. These values apply for clean coils, the results having been corrected for the increase in temperature difference due to scale formation. When operating by the constant-pressure method these changes in salinity would have an enormous effect upon the capacity, but in the orifice method the temperatures and pressures are automatically adjusted to maintain the constant capacity regardless of salinity.

A second series of runs was made at the same capacity as the first series, but with a constant salinity of 3/32nd in both effects, using continuous blow-down. During this run an average total capacity of 108 gallons per day per square foot of coil surface was obtained for a period of 30 hours, during which time the first-effect coil pressure increased slowly from 44.0 pounds per square inch gage to 160.0 pounds, and the second-effect coil pressure increased from 20.4 pounds to 75.0 pounds. The average rate of increase in temperature difference for the first effect was 1 degree per hour and for the second effect was 1.6 degrees per hour. At the end of this period the scale on both effects was cracked by suddenly admitting steam to the coils which were surrounded by cold water. Operation was immediately resumed, after filling the shell with brine of 3/32nd salinity without opening the shell,

and the coil pressures were found to have been reduced to 88.1 pounds on the first effect and 42.2 pounds on the second effect. The pressures then gradually increased at the same rate as previously.

A third series of runs was made with a smaller orifice installed, in order to determine the action at a lower capacity. This orifice was 0.258 inch in diameter and discharged 680 pounds of steam per hour with 250 pounds line pressure. The start was made with clean coils which produced a pressure of 25 pounds on the first-effect coils and 15 pounds on the second-effect coils. Operation continued for a period of 91 hours during which the coil pressure slowly increased to 165 pounds on the first effect and 55 pounds on the second effect. The temperature difference on the first effect increased at a rate of 0.6 degree per hour and on the second effect the increase was 0.5 degree per hour. In each series it was noticed that the rate of increase in temperature difference was smaller during the latter part of the run when scale had formed on the coils and the coil pressures were relatively high, than during the first part of the run when the coil pressures were lower. This disproves a prevalent idea that high coil pressure should not be used on account of unfavorable effects of high coil pressures upon scale formation.

The capacity obtained averaged 64.5 gallons per day per square foot of surface, which is above the present rating of evaporators. The capacity was slightly higher at the start than at the end of the run on account of changes in efficiency due to coil drain losses.

The scale was cracked by the sudden application of steam when the coils were surrounded by cold water. The appearance of the second-effect coils after cracking scale is shown by the photograph, figure 8. It was feared that possibly the scale cracking would not be successful on account of the larger accumulation of scale as indicated by the low heat-transfer coefficient. On the second-effect coils a large portion of the scale

was entirely removed, and most of the scale remaining was in a very loose condition and could be removed with the finger tips or with very little effort. Some of the scale, though loosened, is held in position on account of the close winding of the coils. The coil at the extreme right, which is not wound so closely as the others, shows a better removal of scale. The first-effect coils were also satisfactorily scaled, as shown in figure 9. There is no doubt that this method of removing scale from this type of coils can be developed on shipboard so that the scale may be removed at the end of each period and the operation continued without the necessity of opening the evaporator door, except at long intervals for inspection. A small door should be provided in the lower part of the shell for the purpose of hauling out the scale which falls off. This method of removing scale is, however, not an essential part of the method of operating with an orifice, for the scale may be removed by hand, if desired, when it has reached a condition which demands it.

The orifice method of operation provides a convenient method of investigating the effect of water level, and also of determining the proper level at which to operate. It must be recognized that the level in the glass does not represent the brine level in the shell on account of the boiling and foaming of the brine in the shell, the level in the shell being always considerably higher than the level as indicated by the glass. In fact, the level in the shell has probably a very indefinite value, varying considerably with vapor pressure, feed temperature, salinity and capacity. However, the level in the glass may be used as an indication of the conditions in the shell. A series of runs was made with a constant salinity of 3/32nds in which the water level was varied from 24 inches to 8 inches in the glass, measured above the center of the lower manifold. A run was made for one hour at each level, by 2-inch increments, and the results are plotted in figure 12. As the level was decreased, the coil pressure, and therefore the temperature

difference, slowly increased, till a level of 12 inches was reached, when the pressure went up much more rapidly. The higher temperature differences indicate that a part of the surface is exposed, because the amount of heat transmitted is the same at all levels. In addition to the higher coil pressures required, there is another very decided objection to a low water level. That is the deposit of a large amount of scale on the upper part of the coils. Figure 11 shows the appearance of the coils after operating only 8 hours at a level of 8 inches. The scale is so heavy as to fill in the spaces between the convolutions of the coils. Contrasting this with figure 7, which shows the nature of the scale formed with a high water level, the advantage of high water level is clearly seen. The excessive formation of scale at low water levels is easily accounted for. When the level is low the boiling causes a large amount of brine to be splashed up against the exposed portion of the coils, and as the brine hits the hot coils the water is evaporated and all the salt which was in the water is deposited on the coils, leaving a heavy scale. When the water level is carried high enough to keep the coils surrounded with water a large proportion of the salt stays in solution and is blown out of the shell with the blow-down.

If, after starting with a low level, the level is raised, the excessive scale formed on the upper parts of the coils reduces the heat-transfer coefficient to such an extent that either the desired capacity cannot be obtained or the coil pressure must be increased. With the usual method of operation with constant coil pressure, on starting with clean coils it is generally necessary to start with a low level in order to prevent priming which would result from the high capacity produced at a high level. A heavy scale rapidly forms over a large portion of the coils, thereby preventing the proper transmission of heat, even should the water level be later raised. One fundamental advantage of the orifice method may be seen in the ability to carry a high water level at all times without the danger of

priming, and the consequent production of thin, uniform scale, which may be easily removed, rather than the production of heavy scale similar to that shown in figure 11.

The only objection to a high water level is the danger of carrying over salt. At all levels up to 24 inches the vapor was free of chlorine, and in any properly designed evaporator the level can be carried so as to cover the coils without priming. The high level keeps all of the heating surface in action, reduces the amount of scale, requires lower pressures and increases the time of running without cracking scale. Also, on account of the larger volume of water in the shell, the blow-down period is increased if intermittent blow-down is used. The conclusion is that the level should be carried so as to submerge all the surface, if this is possible, without making chlorine in the vapor. The level at which to operate any evaporator equipped with an orifice may be quickly determined by a short series of runs similar to that plotted in figure 12. In determining the proper level in any specific case it will be well to first try operation with a level in the glass which is 4 or 5 inches below the top of the evaporating surface. If chlorine is produced in the vapor with this level the level must be lowered. The coil pressure should be observed as the level is lowered, and it should be possible to establish a working level at a point which does not produce a greatly increased coil pressure. In most evaporators there will be a considerable range in level over which the coil pressure is nearly constant and the vapor is sufficiently free from chlorine. The working level should be established midway in this range of levels.

COMPARISON OF ORIFICE METHOD WITH USUAL METHOD.

Figure 13 has been prepared to show the comparison between the action of the usual constant coil-pressure method and the improved-orifice method of operation for a single-effect plant. The shell pressure is assumed to be atmospheric and the line pressure is 150 pounds gage. The left-hand ordinates are steam flow through orifice or steam used, pounds per

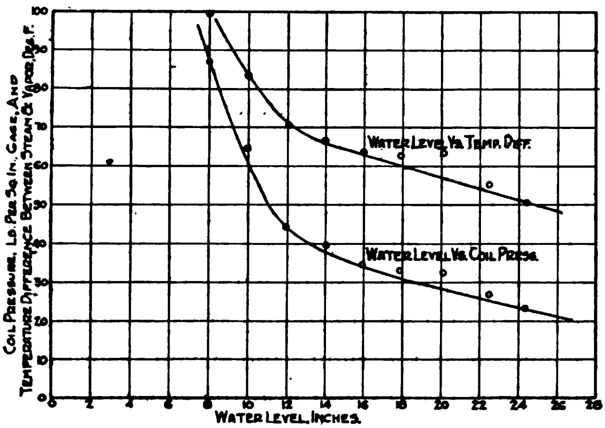

Water Level Vs. Temp. Diff.
Water Level Vs. Coil Press.
Water Level, Inches.
Coil Pressure, Lb. Per Sq. In. Gage, and Temperature Difference Between Steam & Vapor, Deg. F.

FIG 12.

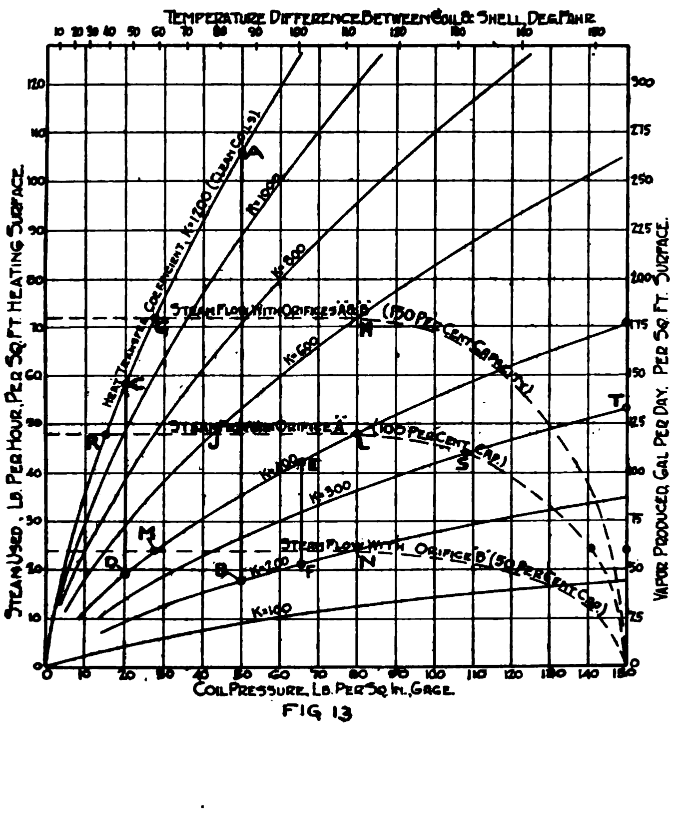

Temperature Difference Between Coil & Shell, Deg. Fahr.
Steam Flow With Orifices A & B (150 Per Cent Capacity)
Steam Flow With Orifice A (100 Per Cent Cap.)
Steam Flow With Orifice B (50 Per Cent Cap.)
Heat Transfer Coefficient, K=1200 (Clean Coils)
K=1000
K=800
K=600
K=400
K=300
K=200
K=100
Coil Pressure, Lb. Per Sq. In. Gage.
Steam Used, Lb. Per Hour, Per Sq. Ft. Heating Surface.
Vapor Produced, Gal. Per Day, Per Sq. Ft. Surface.

FIG 13

hour per square foot of surface, and the right ordinates are vapor produced, gallons per day per square foot of surface. This ratio of steam to vapor corresponds to 0.86 pound of vapor per pound of steam, a fair value for single-effect operation. The solid-line group of curves, diverging from the left-hand origin, are lines of constant heat-transfer coefficient and show the relation between coil pressure and capacity for various values of K. These curves were computed from the heat-transfer coefficient formula. The dotted lines, which extend horizontally till a coil pressure of 81 pounds is reached, and then converge at the right-hand origin, represent steam flow through three orifices and also the capacity corresponding to these steam flows. The horizontal section of the steam-flow curves are computed from Napier's equation, and the curved portion, from the critical pressure of 81 pounds to 150 pounds, is computed from a thermodynamic formula. Orifice B is of such size as to discharge sufficient steam to produce a capacity of 60 gallons per day per square foot of surface. Orifice A is of such size as to produce a capacity of 120 gallons per day per square foot of surface. Orifice A and B arranged in parallel will produce 180 gallons per day per square foot of surface. For convenience orifice A is assumed to produce 100 per cent rating, though this is double the present Navy rating. Orifice B alone produces 50 per cent rating, and orifices A and B combined produce 150 per cent rating.

It must be clearly recognized that every condition of coil as regards scale produces a definite heat-transfer coefficient at which the evaporator must operate. With the constant-coil pressure method of operating, if the attempt is made to start with clean coils and a coil pressure of 50 pounds gage, the operation will be at the point A, on the curve $K = 1,200$, where the capacity would be 264 gallons per day per square foot of surface. With most evaporators violent priming would occur at this capacity, and in order to produce pure water the water level in the shell would usually be lowered till the ef-

fective heat-transfer coefficient was low enough (due to the reduction in surface) to produce a safe capacity. Operation at this low level soon produces an enormous quantity of scale over the exposed portion of the coils. The operation then continues down a vertical line at constant pressure and decreasing capacity as the heat-transfer coefficient decreases to some point such as B, which is for the value of K = 200. Some operators start the operation at a lower pressure, say 20 pounds, in which case the coils may be submerged and the start would be made at C, within the safe capacity of the evaporator. The operation then proceeds down a vertical line to some point D, when the pressure may be increased to say 66 pounds. The operation would continue at E, at the same value of K as existed for the lower pressure, and from E the capacity would again decrease as the operation proceeded at constant pressure and decreasing heat-transfer coefficient.

With an orifice installed, operation always starts at the intersection of the steam-flow curve for the orifice, and the heat-transfer coefficient curve of the coils. For example, with orifice A installed, and with clean coils, operation would start at the point R, and the coil pressure would be 15 pounds. Operation will then proceed in a horizontal line at a constant capacity, instead of in a vertical line at constant pressure. As the value of K decreases the capacity remains constant and the coil pressure increases, instead of the capacity falling off, as in the present method. With orifice A installed, the evaporator will operate at its rated capacity till a coil pressure of 81 pounds is reached at the point L, where the value of K is 400. At this point there is a choice of several things which may be done. The coils may be scaled and the operation resumed with clean coils. (This is the point at which the coils are scaled in ordinary operation.) Or, the operation may continue down along the steam-flow curve, at increasing coil pressures and only slightly decreasing capacities, to some point such as S, where the coil pressure is 110 pounds, K is 300,

and the capacity is reduced only 10 per cent below the rated capacity. If there is no objection for structural reasons to carrying full-line pressure in the coils and steam header, the by-pass valve may be opened wide, allowing the evaporator to operate with full-line pressure of 150 pounds on the coils. This will cause the operation to move up the line of $K = 300$, bringing the operation to the point T, at which the capacity is only 10 per cent above the rated capacity and where the operation will therefore be safe and without danger of priming. From T, the capacity will fall at constant coil pressure as the scale further accumulates. If the coil pressure may not be increased above 81 pounds, and it is desired to continue running when this pressure is reached, orifice A may be closed and orifice B opened. This will mean dropping back along the line of $K = 400$ to the point M, where the coil pressure is only 27 pounds, and 50 per cent capacity is obtained. Operation may now continue at 50 per cent capacity along the line M N till the value of K is reduced to 200 at a coil pressure of 81 pounds.

When it is desired to force the evaporator orifices A and B may be operated in parallel at the start. This will result in starting at the point G with 150 per cent of rating and continuing to the point H where $K = 600$. From this point the operation may be either continued along the steam-flow curve at increasing coil pressures and slightly decreasing capacities, or the operation may be immediately resumed at the rated capacity by closing off orifice B, and dropping back to the point J and continuing at rated capacity.

ARRANGEMENT OF ORIFICE INSTALLATION.

The proposed arrangement of the steam piping to the first-effect coils for orifice operation is shown by the sketch, figure 14. The three-branch construction of standard fittings is substituted for the reducing valve at the coil entrance. The upper branch contains the orifice plate "A" for usual operation at

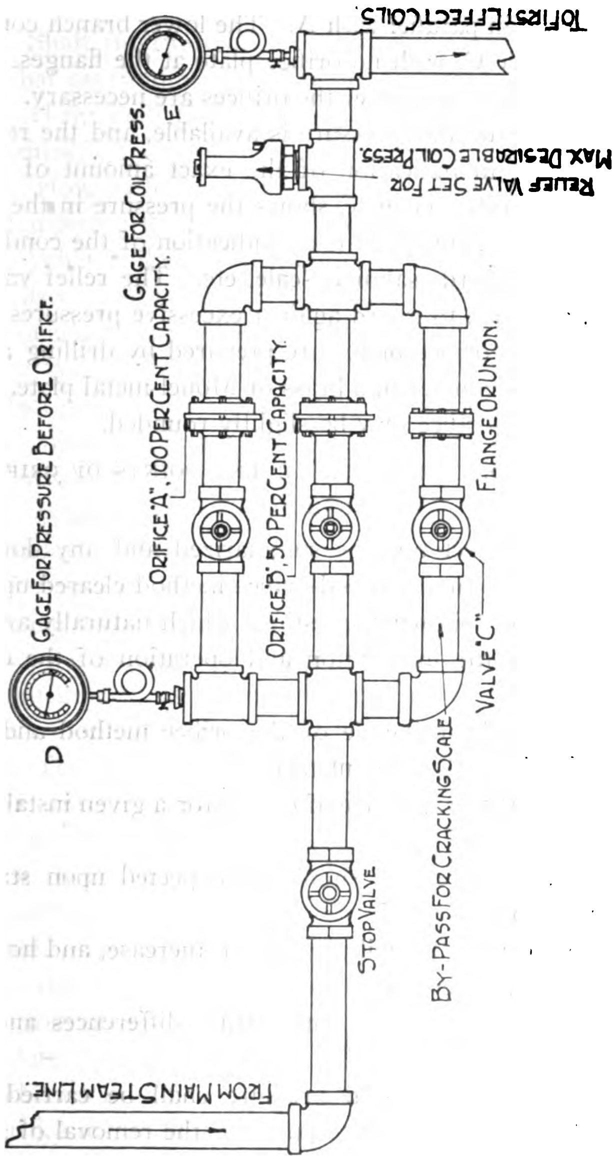


FIG. 14.—SKETCH OF ORIFICE INSTALLATION.

100 per cent capacity. The center branch contains the orifice plate "B" for operation at 50 per cent capacity, or at 150 per cent when used in parallel with A. The lower branch contains the by-pass valve C, with no orifice plate at the flanges. The gages shown before and after the orifices are necessary. Gage D shows if the full-line pressure is available, and the reading of this gage is an indication of the exact amount of steam flowing to the coils. Gage E shows the pressure in the coils, and the reading of this gage is an indication of the conditions in the shell as regards salinity, scale, etc. The relief valve is necessary, as usual, to guard against excessive pressures from any cause. The orifice plates are prepared by drilling a hole of the required diameter in a brass or Monel-metal plate. The inlet edge of the orifice may be slightly rounded.

SUMMARIZED DISCUSSION OF SALIENT POINTS OF ORIFICE METHOD.

The entire subject may be summarized and any doubtful points concerning the improved-orifice method cleared up by a discussion of the following questions which naturally arise in connection with the installation and operation of the orifice method:

(*a*) What is the principle of the orifice method and how does it differ from the present method?

(*b*) How is the proper size of orifice for a given installation determined?

(*c*) What coil pressure should be expected upon starting operation with clean coils?

(*d*) What causes the coil pressure to increase, and how are the coil pressures decreased?

(*e*) At what rate do the temperature differences and coil pressures increase?

(*f*) What maximum coil pressure shall be carried, and when shall the operation be stopped for the removal of scale?

(*g*) What method shall be used in removing scale so that the operation may be continued without opening up the shell?

(*h*) What vapor pressure shall be carried on the lowest-effect shell?

(*i*) Shall intermittent or continuous blow-down be used, and what maximum salinity shall be carried in the shell?

(*j*) How shall the proper water level be determined and maintained?

(*k*) How shall the feed be heated, and what is the effect of feed temperature on capacity and efficiency?

(*l*) What is the best method of removing condensate from the coils to insure that no steam is wasted and that the coils are properly cleared of condensate and air?

(*m*) What is the application of the orifice method to multiple-effect plants, operating on a condenser and with a vacuum?

(*n*) What is the application of the method to evaporators using auxiliary exhaust?

(*o*) What are the principle advantages of the orifice method of evaporator operation?

(*a*) *Principle of orifice method.*—The manner in which the orifice method produces a constant capacity may be explained briefly as follows. The orifice at the coil entrance produces a constant flow of steam into the coils. The water seal or trap at the coil exit insures that only condensed steam leaves the coils. The pressure in the coils automatically adjusts itself to the value required to produce a temperature difference which will force through the coil all the heat required for the condensation of the steam. With no scale on the coils the coil pressure will be low, and as scale forms the pressure will increase. This increase in coil pressure, however, does not reduce the steam flow through the orifice until the coil pressure (absolute) is increased to above 0.58 of the line pressure (absolute). In the usual method of operation the coil pressure is held constant and the capacity decreases as scale forms. In the orifice method the capacity is maintained constant and the coil pressure increases as scale forms.

(*b*) *Size of orifice required.*—The size of the orifice required is determined from the capacity desired and the pressure

of steam available. The capacity may be taken as the maximum capacity which the evaporator will produce without priming, or it may be based upon a certain capacity desired per square foot of surface. The steam required is computed from the known steam economy of the plant, pounds of steam required per pound of vapor produced, or from the expected economy, considering number of effects, feed temperature, blow-down losses, etc. For single-effect plants the steam economy will vary from 1.20 to 1.35 pounds of steam per pound of vapor, and for double-effect plants the steam economy will vary from 0.60 to 0.75 pound of steam per pound of vapor. If available, the high-pressure steam line should be used. The size of orifice for producing 100 per cent capacity is then computed from Napier's formula,

$$W = \frac{P\,A}{70},$$

in which,

W = steam required, pounds per second.

P = steam pressure, pounds per square inch, absolute, before orifice.

A = area of orifice, square inches.

It is recommended that an additional orifice of 50 per cent capacity be installed and that the two orifices and a by-pass for the purpose of cracking scale, be installed in parallel as shown in the sketch, figure 14. No reducing valve is required, because in the orifice method, the coil pressures are automatically regulated. The gages are necessary for the intelligent observation of the performance. The gage at the orifice entrance measures the steam flow and the gage at the coil entrance indicates the conditions in the shell as regards scale, etc. The relief valve is required, as usual, which should be set for the highest allowable working pressure of the coils. If the orifice installed does not produce the capacity desired, on account of some condition which was unknown when computing the size of the orifice, the diameter of the orifice may

be easily increased by reaming till the desired capacity is obtained. The thickness of the plate used should be about 1/8 or 3/16 inch.

(*c*) *Initial coil pressure.*—The coil pressure which should be expected upon starting with clean coils may be determined from a consideration of the heat-transfer coefficient formula,

$$K=\frac{(H_s-q_d)W}{S\times T.\ D.},\ \text{or transposing,}\quad T.D.=\frac{(H_s-q_d)W}{S\times K}.$$

The value of (H_s-q_d), heat removed by the condensation of steam in the coils, B.t.u. per pound, is nearly 1,000. The value of the heat-transfer coefficient K, is also approximately 1,000 for clean coils and 3/32nds salinity. Substituting,

$$T.D.=\frac{W\times 1{,}000}{S\times 1{,}000}\ \text{or}\ T.\ D.=\frac{W}{S}.$$

The temperature difference to be expected on starting with clean coils is therefore approximately equal to the pounds of steam per hour per square foot of surface. The coil pressure corresponding to the temperature difference is determined from the principle that the pressures in the shell and coils correspond to the temperatures.

(*d*) *Increase in coil pressure.*—The main factor causing the gradual increase in coil pressure is the gradual accumulation of scale on the coils. The coil pressure is also increased by increase in vapor pressure, increase in salinity of brine, low water level, and air binding in the coils. When the coil pressure has increased to its maximum value, attention should first be given to the drain pot to see that the coils are properly drained of air and condensate. Then it must be ascertained whether the water level is at the proper height. The salinity should be taken, and if above 3/32nds it should be reduced by blowing down. If the coil pressure remains high after giving proper attention to these points the scale must be removed from the coils either by hand or by the scaling method described herein.

(*e*) *Rate of increase in temperatures and pressures.*—The tests showed an increase of temperature difference of 1 to 1½ degrees per hour at a capacity of 110 gallons per day per square foot of surface. At a capacity of 64.5 gallons per day per square foot of surface the rate of increase was ½ degree per hour. In every run it was noticed that the rate of increase was smaller toward the end of the run, when the pressure was high, than at the start. In operating with the orifice method a careful log should be kept of the increase in pressure and temperature, and every effort should be made to adopt methods of operation which will keep this rate of increase of temperature difference as small as possible, as the increase in temperature determines definitely the length of time a desired capacity may be maintained. The rate of increase of temperature difference is influenced by probably many conditions of design and operation, and further developments in the orifice method will be along lines of obtaining the smallest possible rates of increase of pressures and temperature differences.

(*f*) *Maximum coil pressures.*—The first-effect coil pressure (absolute) may be allowed to increase to 0.58 of the line pressure without any decrease in steam flow. The maximum coil pressure is limited only by the safe working pressure of the coils and steam manifolds. For these reasons it is advisable to use the highest line pressure available for the initial pressure at the orifice entrance. If 150 pounds line pressure is used, the coil pressure in the highest effect may increase to 81 pounds without decrease in steam flow, but with 250 pounds line pressure, the coil pressure may be allowed to increase to 140 pounds (if this is a safe working pressure of the coils) without decrease in steam flow. The use of the highest available pressure simply means that the operation at any given capacity may be continued to lower values of heat-transfer coefficient, and therefore for a longer period without cracking scale. With the usual method, high coil pressures are not

advised on account of the danger of priming, but with the orifice method, where the capacity is fixed, this danger does not exist, and there appears to be no objection to allowing the operation to continue until the coil pressure has increased to the allowable working pressure of the coils. In double-effect plants the second-effect coil pressure must not be allowed to increase above the maximum allowable first-effect shell pressure.

From figure 13 it is seen that the heat-transfer coefficient which exists when the operation must be stopped on account of reaching a maximum pressure depends upon the capacity at which the evaporator is being operated. Assuming a maximum allowable coil pressure of 80 pounds gage, at 150 per cent rating the limit of operation occurs when the heat-transfer coefficient is 600. At 100 per cent rating, the limit of operation is at a value of K = 400 and at 50 per cent of rating (60 gallons per day per square foot of surface) the operation need not cease till the value of K has been reduced to 200. Thus, at the lower rates of evaporation the operation may be continued much longer.

(*g*) *Removal of scale.*—When the coil pressures have increased to the maximum permissible values at any desired rate of evaporation the scale must be removed from the coils either by hand or by a self-scaling method. The orifice method of operation permits of the carrying of a high water level, and this results in the production of a uniform and relatively thin coating of scale over the entire surface of the coils. It has been repeatedly demonstrated that this sort of scale may be effectively removed from coils by the self-scaling method described herein. The method consists of covering the coils with cold water and suddenly admitting a steam pressure of 50 or 60 pounds on the coils through a by-pass steam connection. The coils must be kept thoroughly clear of condensate and air during this operation by watching the drain pot carefully. This is necessary in order to bring up the tem-

perature of the metal of the coils as rapidly as possible. As soon as the water in the shell becomes warm the steam should be turned off the coils, to prevent the evaporation of the water at a high rate and the carrying over of salt. In some cases it may be necesary to repeat the operation to effect a thorough cleaning. The principle of the method is the difference in expansion of the metal of the coils and the scale. Although this method of cracking scale has been demonstrated as effective only on coils of the types shown in the photographs, it may possibly be successfully applied to other types of evaporators with a little development.

The coil pressure obtained upon resuming operation after cracking scale is an indication of the thoroughness with which the scale has been removed. A sludge door should be provided for hauling out the scale which is cracked off. This door is also convenient for washing off the coils with a hose and for inspection of coils.

(*h*) *Shell Pressures.*—At present it is sometimes considered necessary to hold a pressure of 5 pounds on the lowest-effect shell to prevent priming. This has probably been necessary to reduce the temperature difference when carrying a constant coil pressure. When using the orifice method the temperature differences adjust themselves, and all of the tests made with the orifice method prove there is no reason for carrying higher than atmospheric pressure in the shell of the lower effect. The valve between the lowest-effect shell and the distiller should be opened wide, and with atmosphere pressure on the distiller, the shell pressure may be slightly above the atmosphere. In a two-effect plant, the first-effect shell pressure will increase slowly and the operation must be stopped when a pressure is reached which is limited by the highest safe working pressure fixed by the strength of the shell. Relief valves should be fitted as usual to prevent exceeding the maximum safe pressure.

(*i*) *Blow-down.*—With sea water as feed the blow-down period may be computed from the formula,

$$P=\frac{V(S-1)}{W},$$

in which,

P = blow-down period, hours.

V = pounds of water in the shell at the operating level.

S = salinity at end of blow-down period, pounds of salt per 32 pounds of water (32nds).

W = vapor produced, pounds per hour.

A maximum salinity of 3/32nds is usually recommended. Certainly the blow-down should never take place at lower salinities than this, on account of the unavoidable loss of heat in the brine blown out. In the orifice method of operation the salinity may, if desired, be carried slightly higher than 3/32nds before blowing down, for the following reason. With the orifice method a high salinity merely increases the coil pressures, whereas in the operation with constant coil pressure, the increase in salinity reduces the capacity on account of the decrease in heat-transfer coefficient. If the capacity and volume of shell are such as to require too frequent a blow-down, as computed from the above formula, intermittent blow-down cannot be recommended. The blow-down should then be continuous, and the salinity regulated by the use of a salinometer. A semi-continuous blow-down consisting of blowing down a certain amount of brine, amounting to 5 or 6 inches in the glass, at regular intervals, may be used as an alternative. There are many instances where evaporators are operated without the use of salinometers, and the salinities are allowed to increase to beyond the saturation point of the brine, with disastrous effects upon scaling of coils, capacity, and purity of vapor. It is absolutely impossible to obtain satisfactory evaporator operation without giving proper attention to this important feature. With the orifice method, on

account of the constant capacity feature, it is possible to compute accurately the proper blow-down period. If intermittent blow-down is possible, it is convenient to combine with the blow-down a partial scale cracking. After the brine has been blown out the shell may be filled to the working level or slightly higher with cold water (instead of the heated feed), and a steam pressure of 50 pounds put on quickly through the by-pass. In a double-effect plant a live steam connection should be made to the second-effect coils for this purpose. As soon as the water in the shell becomes warm the by-pass should be closed, and operation continued with the orifice. It will be found that this will produce at least a partial cracking of the scale.

(*j*) *Water Level.*—In the present constant-coil pressure method of operation it is frequently impossible to commence the operation with a water level sufficiently high to cover the coils. A low level must be carried to avoid priming, with the result that the upper portion of the coils are quickly covered with a heavy scale, the capacity is decreased, and the removal of scale is made difficult. In the orifice method the proper level may be easily determined and carried at all times. In most types of evaporators a level 4 or 5 inches in the glass below the top of the evaporating surface will cover the surface with brine and be safe against priming. A definite rule for the water level cannot be given, because the proper level will be different in different designs of evaporators and also at different capacities and vapor pressures in the same evaporator. A large capacity, on account of increased bubbling and foaming in the shell, will possibly require a lower level in the glass than smaller capacities, while a higher vapor pressure will allow a higher level to be carried. The point is that with the orifice method a certain proper level for a given capacity may be easily determined, and this level will be such as to cover the coils, and thereby reduce enormously the formation of scale.

(*k*) *Feed Heating.*—In the orifice method the steam flow is constant, but the efficiency, and therefore the quantity of vapor produced, depends upon the temperature of the feed. Inasmuch as the vapor produced from the last effect must be condensed, there appears to be no excuse for not using a part of this vapor in heating the feed to as high a temperature as possible. The best arrangement for heating the feed is unquestionably that recommended by Commander Dinger,* which consists of: " first allowing the feed to pass through the vapor heater, where it may be heated up to the temperature of the vapor, about 210 degrees F., and then to cause the feed to the first-effect shells to pass through a first-effect coil drain heater where it can be heated to the temperature of about 270 degrees F. In a double-effect plant, with this arrangement, the feed water can enter the evaporator shells at about 210 degrees for the second effect and 270 degrees for the first effect, where the initial pressure in the H.P. coils is 50 pounds." It would be very desirable to have the vapor heater arranged in series with the distiller condenser, so that all the vapor from the second effect passes through the heater. Adequate heating of feed saves not only in reducing the heat required for producing vapor, but reduces enormously the blow-down losses.

(*l*) *Removal of condensate.*—Either a trap or a drain pot may be used to remove the condensate from the coils. In either case it is absolutely essential to have a water glass which records the level of the water seal and which is in plain sight at all times to insure that no steam is blowing through. A water glass on the steam manifold or steam header of the evaporator is sometimes used, but this is wholly unsatisfactory. The condensate must be drawn off into a separate vessel, consisting of either a trap or drain pot, to insure that the coils are free from condensate. If a drain heater is used it should be located beyond the drain pot. If a heater should be included as part of the drain pot, live steam would be drawn from the

*" Marine Feed Water Heating," A. S. N. E. JOURNAL Feb., 1914, P. 158.

coils to heat the feed and therefore no advantage would be gained. A pressure must be maintained on the drain heater in order to heat the feed above 212 degrees. This pressure is most easily obtained by regulating the drain-pot water level from a point beyond the heater. *The air vent on the drain pot should be provided and used.* It is just as necessary to remove the air which collects in the coils as a result of the condensation of the steam as it is to remove the condensate. If the coils become filled with air, or air bind, the heat cannot be transmitted through the coils and excessive coil pressures will result.

In designing the drain pot it should be noted that the larger the horizontal sectional area of the drain pot the easier will be the control of the level in the glass. A number of drain pots have been seen whose unsatisfactory action has been due to their relatively small sectional area.

(*m*) *Multiple effect application.*—Three or four-effect operation with the lowest effect operated at a high vacuum is at present very troublesome because of the difficulty of preventing priming in the lower effects. This is largely due to the following condition. At the lower pressure the temperature difference due to a small pressure difference is comparatively large, and it is almost impossible to know at what pressures to operate to produce a given capacity without the danger of priming. Also, a small change in vacuum produces a considerable change in temperature difference which induces priming. At the high vacuua at which the lower effects operate the specific volume of the steam is very high, and on account of the consequent high velocities of vapor at the disengaging surface it is to be expected that priming will occur at relatively low capacities, unless the disengaging surface and baffling have been especially designed for the high vacuum operation.

Having established the maximum capacity which the lower effects may produce without priming, the orifice which sup-

plies steam to the first-effect coil may be designed for a steam flow which will produce this capacity. The air must be removed from the coils which are under a vacuum by leading a small connection from the top of the drain pot or trap to the condenser.

(*n*) *Application to operation with exhaust steam.*—The orifice method may also be applied to advantage to the operation of evaporators with auxiliary exhaust. The flow of steam through an orifice operated on exhaust steam at a pressure of 15 pounds gage will be constant in amount as long as the pressure on the discharge side is 2.5 pounds gage or less. With this for the maximum coil pressure, the shell will need to be operated under a vacuum. With a vacuum of say 26 inches, a temperature difference of 100 degrees is available. With clean coils this temperature difference is not required, and in single effect operation would cause serious priming when operating by the usual method. With an orifice installed, the capacity will remain constant regardless of vacuum or temperature difference until the coil pressure reaches 2.5 pounds gage. The operation need not be stopped here, but can be continued until the coil pressure has increased to 3 or 4 pounds gage with only a slight decrease in capacity. At this point the by-pass may be opened and the operation continued with the full exhaust pressure of 15 pounds on the coils without producing excess capacity with the consequent dangers of priming. The value of the method is that upon starting with clean coils a vacuum may be carried on the shell and the proper water level may be carried without danger of the priming which would otherwise result. In fact, all the inherent advantages of the orifice method are realized.

(*o*) *Advantages of method.*—The advantages of the orifice method may be summarized as follows:

(*a*) Any predetermined capacity, within the limit of capacity of the evaporator, may be obtained and maintained.

(*b*) The liability of priming is reduced to a minimum.

(*c*) Higher water levels may be safely carried at all times without danger of priming, and, for this reason,

(*d*) The scale forms on the coils in a thin uniform layer, and may be easily removed.

(*e*) The best operating conditions may be easily determined, and, on account of the constant capacity, may be easily maintained. At the same time, variations from the desired operating conditions, within wide limits, do not affect the capacity.

(*f*) The reducing valve is dispensed with. The evaporator is operated directly on the line pressure, and no pressure regulation is required. There is no danger of sudden increases of coil or shell pressures.

(*g*) The pressure on the inlet side of the orifice indicates exactly the amount of steam being used by the evaporator. The pressure on the coils indicates the condition of the coils as regards scale and heat-transfer coefficient.

(*h*) The system may be easily applied to any existing evaporating plant, single or multiple effect, high or low pressure.

It is believed that the adaption of the orifice method to the operation of evaporators will remove from the evaporating plant much of the mystery and uncertainty with which its operation is at present associated, and will place the evaporators upon as safe and sound a basis as have any of the ship's auxiliaries.

Acknowledgments are due to Rear Admirals C. W. Dyson and T. W. Kinkaid and to Mr. W. L. DeBaufre for encouragement and suggestions, and to Mr. Leslie E. Sebald for valuable assistance in the conduct of tests and the preparation of this article.

LUBRICATION AND LUBRICANTS.

By G. R. Rowland.*

"Lubrication" and "Lubricants" may be defined as follows: Good lubricants are substances, semi-fluid or fluid, capable of forming and maintaining films of sufficient thickness between two rubbing surfaces to prevent actual friction between the surfaces, substituting for it the fluid friction of the lubricant itself.

Lubricants in marine practice may be classed as mineral oils, animal oils, vegetable oils and greases.

Mineral oils are manufactured from crude petroleum and shale. The petroleum lubricating oils used in the United States are made from crude oils which may be classed in the following manner according to their natural characteristics:

Paraffine Oils—Composed chiefly of hydro-carbons of the paraffine series and found chiefly in Pennsylvania, West Virginia and Eastern Ohio.

Asphaltic Oils—Composed of hydro-carbons of the napthene series, found principally in California, Texas, Louisiana, and the Mid-Continent Oils produced in Kansas, Oklahoma and Illinois.

Lubricating oils made from the paraffine crudes have high gravities, high flash and fire tests, high congealing points and low viscosities unless blended with steam-cylinder oils. Asphaltic base oils have low gravities, lower flash and fire tests, congeal only at temperatures below zero degrees F., and can be made in any desired body without blending.

Lubrication can perhaps be best understood by considering the case of a spindle revolving at a speed of 10,000 revolutions

*Supervising Engineer, The Texas Company.

per minute in a bath of oil. The oil film immediately touching the surface of the spindle will revolve in the bath at nearly the surface speed of the spindle, the oil clinging to the spindle through its power of adhesiveness. The next succeeding films or layers of oil revolve at slightly less speed, while the film touching the sides of the bath will be as nearly stationary as its cohesiveness and the general movement of the oil will allow. The friction of the particles of oil sliding past one another is referred to as internal or fluid friction.

If a gradual side pressure is applied to the spindle, forcing it towards one side of the bath, some of the oil will be pressed out, and when the pressure becomes great enough the metal surfaces of the spindle and side of the bath or base will actually come together. This will produce heat and wear from metal friction. If the pressure is constantly kept at the point where the metal surfaces actually come together, an oil having more body or viscosity should be used. The heavier-bodied oil will resist the pressure and keep the surfaces apart, and the spindle will again turn in the center of the bath. As long as the oil is heavy enough to keep the metal apart no metal wear takes place. Again increasing the side pressure on the spindle will force the heavier oil out, and heating will again occur. To overcome a steady pressure of this nature a still heavier-bodied oil must be used in order to keep the surfaces apart.

In the above example the frictional heat of the spindle base will be slightly increased as the heavier oil is used under heavier pressure, but in no case will the heat be as high as that resulting from the destructive abrasion of metal on metal due to the use of too light an oil. Also, the minimum amount of power will be required to operate the spindle with the lightest oil in use, and the power required will slightly increase as the heavier oil is used; but, as in the case of temperature, the power will at no time be as high as when the surfaces are together through too great pressure or too light an oil. This case illustrates the effect of pressure upon the oil film and upon

the viscosity or body of the oil. The same rules apply generally to all bearing lubrication.

Lubricating oils can be divided into two general classes. The first class, called engine and machine oils, including dynamo, turbine, spindle oils, etc., is intended for external lubrication. The second class, steam-cylinder and valve oils, is used for the internal lubrication of steam cylinders and valves. Motor oils and oils for internal-combustion engines may be considered as a separate class, though they are not essentially different from some of the highest class turbine, engine and machine oils.

One cause of loss of power in a steam engine is friction. In all engines there are so many moving parts that it is of the greatest importance that friction should be reduced as much as possible. This is done by making the surfaces in contact smooth and of ample size; also making them of different metals and the use of suitable lubricants. Most of the marine engines in use at the present time are vertical multiple-cylinder machines built without fly-wheels or governors. Up until about twenty years ago the moving parts of marine engines were lubricated with various kinds of animal oils, principally lard oil. Of late years, or since the introduction of mineral oils, animal oils have been almost completely eliminated due to the fact there has been a vast improvement in mineral oils and also that lubrication is today done on a more efficient basis. In most all reciprocating engines the oil is reclaimed, filtered and used over again, which was impossible to accomplish to any great extent with animal oils.

Engine and machine oils should meet the following requirements:

(1) The oil should be of sufficient viscosity or body to keep the bearing surfaces apart at working temperatures.

(2) It should possess such qualities as will reduce frictional losses to a minimum.

(3) It should remain fluid at such low temperatures as will be met under ordinary service conditions.

(4) It should meet service requirements as to durability, separation from water, etc.

(5) The flash point should be sufficiently above the working temperature to allow a reasonable margin of safety.

(6) It should have no tendency to decompose or to form deposits which will gum up the machine and increase the friction.

(7) It should contain no impurities which will corrode or pit the bearing metal.

In general, but more especially with high and medium-speed machines, the best practical guide as to the suitability of a lubricating oil is the amount of frictional heat produced with that oil in use. Frictional heat indicates wasted power, and, in the case of a bearing, is determined by the difference between the temperature of the bearing in operation and the temperature of the room in which the machine is running. In general, one of the indications of an improvement in lubrication is the reduction of the amount of frictional heat. However, there are so many other factors which enter into the determination of the ultimate value of a lubricant that the amount of frictional heat as indicated by the bearing temperature is not in itself conclusive proof of the superiority of one lubricant over another.

However, the military safety of a battleship is of greater importance than a slight reduction in the temperature of a turbine bearing. A hot bearing or a burnt-out bearing at a critical moment might mean the loss of the ship, whereas the undue heating could have been prevented by the use of an oil of sufficiently high viscosity. In the case of Diesel engine lubrication, due to the fact that the unused cylinder oil drains out and mixes with the oil in the general lubricating system, it is desirable to use one oil sufficiently heavy for use in the cylinders, though this oil might cause additional internal friction in the bearings.

Motor oils and oils for internal-combustion engines are sub-

jected to extraordinary conditions. Their requirements are therefore somewhat different from other lubricating oils, and may be classed as follows:

(1) The viscosity should be such that the film of oil will be sufficient to keep the metal surfaces apart.

(2) The oil should possess such characteristics as will permit it to form a seal between the piston rings and cylinder walls so as to secure a high degree of compression.

(3) It should be of such a nature as to be converted into vapor without decomposition, even under extreme heat, so that the oil which gets into the combustion chamber will pass off with the exhaust as vapor without leaving an excessive carbon deposit.

(4) It should be of such a nature that whatever carbon is deposited is soft, and easily removed.

(5) The cold test should be low enough to meet all temperature conditions.

The proper viscosity for a motor oil should be determined by the piston-ring clearance and the conditions of the rings and cylinder walls. A wide clearance or worn rings, either of which causes poor compression, requires a high viscosity oil in order to secure the best results. If the piston rings are closely fitted to the cylinders a light-bodied oil should be used. If too light an oil is used too much oil will get past the piston rings on the suction stroke, and on the compression stroke some of the fuel mixture will leak past the piston rings and condense in the crank case, making the oil still thinner. If the oil is too viscous there will be a loss of power. Good compression depends very largely upon proper lubrication.

Mechanical conditions involved in steam-cylinder lubrication require special oils for that purpose. In color cylinder oils are usually dark green or black, though the filtered oils are light green and bright red. Steam cylinder oils are, as a rule, very viscous, some of them scarcely flowing at ordinary temperatures. The viscosity of these oils is taken at 210

degrees F., whereas that of ordinary lubricating oils is usually taken at 100 degrees F.

The conditions which are to be met in the steam cylinder are quite different from those occurring in the lubrication of external parts. In the case of steam-cylinder lubrication the oil is broken up or atomized and used only once and is then carried off with the exhaust steam. The chief considerations in the selection of a steam-cylinder oil should be, first, its viscosity, and, second, whether or not it should be compounded.

The speed and type of the engine, the method of effecting the lubrication, the distance from the steam chest to the point where the lubricant enters the steam line, and the amount of moisture in the steam should all be taken into consideration in determining the proper viscosity for a given case. It is necessary that the oil should be atomized or broken up into minute parts before it reaches the valves so that it will be carried by the steam and spread over the valves and cylinder walls in a finely-divided state. Otherwise, the oil will pass out of the exhaust in nearly the same condition in which it entered the steam pipe, without providing any effective internal lubrication. A light-viscosity oil atomizes more readily than a heavy-viscosity oil, so that the greater the amount of moisture in the steam the lower the viscosity of the oil should be. The shorter the distance which the oil has to travel in the steam pipe, the less the opportunity for atomization, and therefore the lower the viscosity of the oil should be. On the other hand, the farther away from the steam chest the oil is put into the steam pipe the heavier it can be, as time and distance are required for the heavier oil to properly atomize.

In ordinary cases, where there is a considerable amount of moisture in the steam, the cylinder oil should be compounded with a certain amount of animal oil so that the oil will more readily form an emulsion and adhere to the cylinder walls. On the other hand, a straight mineral cylinder oil should be used where it is desirable to keep the animal matter out of the

exhaust steam, as for example when the steam is condensed and returned to the boilers and used for other purposes.

FIXED OILS AND COMPOUNDED OILS.

Before the introduction of mineral oils for steam cylinders tallow was used exclusively, being melted in a tallow pot which was kept on top of the steam chest. After the tallow was properly melted, it was introduced into the cylinder through a tallow cup, usually fitted with a valve and screw cover, and located on top of the steam chest. Lubrication by this method was crude and very intermittent. However, it probably suited the times and the condition of the engines, examples of which can be seen and studied with interest in some of our great museums. Sperm oil was formerly used, both mixed and unmixed, with mineral oils, for light machinery and spindles. Porpoise-jaw oil is still used for the lubrication of watches and delicate mechanism, for which a thoroughly non-drying and fairly limpid oil having an extremely long life is required.

As far as the lubricating properties of animal and vegetable oils are concerned, they were excellent, as the heavier oils, such as castor, were able to withstand very high pressure, and the lighter oils, such as sperm, had a very low coefficient of friction. The great drawback to their use, in addition to their high cost, was their oxidizing tendency, which necessitated the frequent cleaning of machinery to get off the gum formed by this oxidation. Moreover, acids were formed, which attacked the metals, causing pitting and corrosion. Therefore, pure mineral lubricating oils, if properly manufactured, can be used over and over again, which is not the case with any of the fixed oils.

Nearly all lubrication of a general nature can now best be cared for by a straight mineral oil of the proper viscosity. For turbines or for any circulating system, a compounded oil is never used, as it causes an emulsion, which above all is to be

avoided, and only the best grades of straight mineral filtered oils should be used for the latter purpose and should be selected strictly upon the basis of quality and suitable viscosity.

GREASES.

In general lubricating practice there are certain mechanical conditions which cannot be handled to advantage by lubricating oils, and for which it is necessary or desirable to use a semi-solid lubricant. To meet such conditions the form of the mineral oil is changed to a semi-solid state by a special process, and the resulting lubricant is known as grease.

Grease, as the term is used for commercial purposes, is a mineral oil thickened with soap. The soaps used for this purpose are made from an animal or vegetable oil, such as tallow oil, horse fat, lard oil, cottonseed oil, rapeseed oil, palm oil and rosin oil. The consistency of the grease is from soft to hard, depending upon the percentage of soap in the mixture. In practically all greases mineral oil constitutes by far the larger part, frequently as much as ninety per cent of the finished product.

A grease manufacturer must be very expert in the methods of handling the various materials used in the manufacture of grease. Two greases might be made according to the same formula and still vary greatly in lubricating value, due to the difference in the methods of handling the various products used in their manufacture.

In the manufacture of most greases the animal or vegetable oil is placed in a steam-jacketed kettle and heated to the required temperature. Cream of lime or other alkali dissolved in water is then added, and this mixture is boiled and stirred until it is completely saponified, or turned into soap. In the manufacture of the best greases the heating is continued until nearly all of the moisture is driven off. Part of the mineral oil is then stirred in, and the stirring is continued with the

addition of more oil until the proper consistency is reached. Lime and soda greases are made by first mixing the fixed oil and mineral oil and then adding the alkali. The finished product should contain no lumps and should be homogeneous in character.

For all lubrication where the speed is high and where oil can be retained in or around the bearing, it is inadvisable to use grease, for the reason that lower bearing temperatures and lower power consumption can be secured by the use of oil. There are many places, however, where it is necessary to use grease, due to the fact that oil cannot be retained on the bearing surfaces, or due to the fact that the bearings are so located that it is impossible to get at them to oil them regularly. It is therefore necessary to have a lubricant that will feed very slowly and insure lubrication for an extended period without replenishing, and as a general rule, grease should be used for lubrication only under slow speeds and high pressures, or in cases where fluid lubricants cannot be retained for a sufficient time or used with sufficient cleanliness during the period of operation.

The melting point of grease is the temperature at which the grease begins to flow in liquid form. There are several methods of determining the melting point and each method will give a different temperature as the melting point. The following method, Fig. 1, is now being used very generally with satisfactory results. Some of the grease to be tested is spread with the fingers onto the bulb *B* of a thermometer, which ranges from about 100 degrees F. to about 400 degrees F. The thermometer is then fitted by means of a cork, *C*, into an empty test tube *T*, so that the thermometer hangs free. The test tube is then placed in an oil bath *O* heated by a Bunsen burner, and the flame so regulated that the temperature will rise from two to four degrees per minute. When a drop forms on the bulb of the thermometer and falls to the

bottom of the test tube, the temperature is recorded as the melting point of the grease.

Cup greases are primarily intended for use in compression cups which force the grease directly to the bearing surfaces. There are, however, a number of designs of compression and other types of cups. As cup greases are comparatively low in melting point, very little frictional heat is required to reduce them to a liquid, consequently their consistency should depend very largely upon the mechanical method used to apply them to the bearing surfaces. The personal equation plays a large part in the selection of greases of various consistencies, as different engineers with the same equipment vary in their opinions, some preferring a very soft and others preferring a very hard grease, irrespective of the requirements of the machine.

Unless compression cups are used, grease must depend upon the frictional heat of the bearing to soften it enough to be carried in between the bearing surfaces. At the same time it is necessary that the grease should not melt and flow away from the bearings, but should stay in a semi-solid state under bearing temperatures. Special grease, intended for use in relatively high temperatures, can be of the same general consistency as the compression-cup greases, but of higher melting point. This permits of direct application to the bearings, as in the case of open-slot bearings, or it can be fed through a copper pin cup. This cup does not have the compression device, but instead has a copper wire, the function of which is to transmit heat from the bearing to the grease, so as to make it fluid enough to flow to the bearing. For extremely high surrounding temperatures a very hard and a very high melting point grease should be used.

If a grease is not properly manufactured, the oil will frequently separate under service conditions, leaving the soap in the cups and on the bearings in solid or powder form. This is caused by a defect in manufacture that cannot be discovered

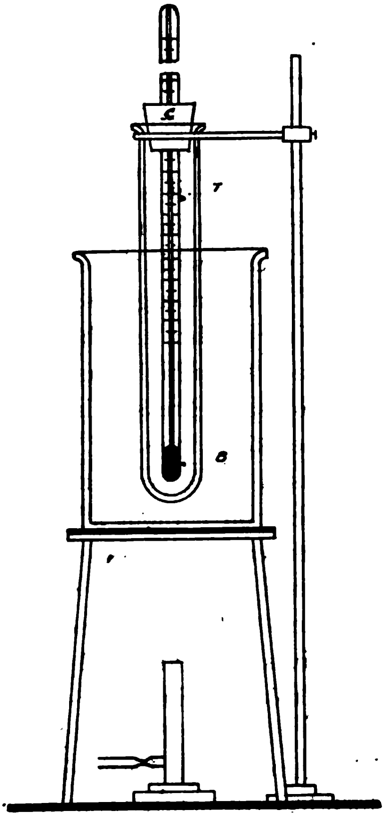


FIG. 1.

FIG. 2.

by the casual inspection of a sample of grease. Cup greases and general lubricating greases should not separate when subjected to working temperatures, and when cooled should go back into a perfectly homogeneous mixture.

A grease is never better than the mineral oil used as its base. In other words, the lubricating value of a properly manufactured grease depends upon the mineral oil used in its manufacture. The soap acts as a thickener for the mineral oil, and it is the latter that does practically all of the lubricating.

The appearance of a grease is frequently very misleading. In endeavoring to make greases at a low price, it is the custom of some manufacturers to use the very cheapest mineral oils obtainable. It is possible to make a grease with these cheap oils which will appear just as good to the eye as a grease made from the very best grades of lubricating oil, but as lubricants they are greatly inferior to greases made from high-class mineral lubricants. In the manufacture of some greases the soap is not boiled down so as to drive off the excess moisture, and as a result the finished product contains a large percentage of water. This lowers the cost of production without affecting the appearance of the grease. Moreover, many greases for heavy pressures and for rough work are filled with powdered soapstone, chalk, talc, and other fillers. This increases the weight, cheapens the product, and gives it an artificial body. Such fillers, however, should never be tolerated in a high-grade grease.

The low temperature to which a grease can be subjected without solidifying is a point which has frequently been overlooked in the past. It is particularly necessary that greases used in low temperatures should be manufactured from low cold-test oils, and therefore they will not become solid even in very severe winter weather.

There are certain conditions which the hardest and the highest melting point greases cannot satisfactorily meet, and include such work as that offered by different types of gears

and wire ropes. To meet these exact conditions a product has been perfected which, due to its great viscosity, its cohesiveness, its adhesiveness, and its heat resisting properties, is now being used with wonderful success under conditions where ordinary greases have always failed to show efficient lubrication. A thinner lubricant has been developed for worm drives and other small enclosed high-speed gears.

These same products are used under conditions as affected by temperatures in connection with wire-rope lubrication. Here it has been found that the ordinary or best grades of oils or greases have only provided a temporary coating of the outer surface of a wire rope. While the outer coating lasted it offered some resistance to friction between the surface of the rope, the sheaves or drums over which the rope passes or to the attacks of the elements to which the rope may be exposed, but it is the friction and corrosion of the inner strands that eats into the life of the cable. However, the lubricant which has recently been developed will, when properly applied, work in between the strands and thoroughly lubricate every wire of the rope as well as the hemp center; it will not harden on the outside of the wire rope and will not peel or crack, and any rope dressing that will harden or crack will let in water, which rusts the wires, and rust is the great enemy to a wire rope.

In order that the engineer may be in a position to more intelligently select suitable lubricants a little general information concerning the separation of the natural crude oils into various products used for lubrication purposes, and also by touching upon the most important test to which lubricating oils are submitted will give a better idea regarding the selection of proper lubricants, it being desired to bring out clearly those qualities which will cause an oil to be suitable or unsuitable for the contemplated work.

OIL REFINING.

Briefly, crude oils are separated into various products by

the following methods of distillation—the "dry" distillation process in which the heat is applied directly to the bottom of the stills and the distillation continued until coke only remains and the "steam" process in which beside the direct heat steam is permitted to enter at the bottom and bubble up through the body of the oil, also the "vacuum" process in which, by means of a pump, a partial vacuum is created in the still. These various methods may be used singly or, as in most cases, two or more are combined before the finished products are arrived at. The advantage or disadvantage of using either or any of them would require too much space to properly cover here.

The stills, of course, vary greatly in design and size. In any case they are similar to an ordinary steam boiler without the tubes, usually being a horizontal steel shell set in brick work and connected at the top or dome with vapor pipes which lead to the condenser coils.

The boiling points of the various constituents of the crude of course determine when that part vaporizes and passes to the condenser.

The more volatile portions of the crude, such as benzine, gasoline, kerosene and gas oil are distilled first, the cut from which general lubricating oils are made coming over considerably later, and the steam-engine cylinder stock is the residue which remains in the still after the other products are drawn off, providing steam is used in distillation.

Each cut coming from the still may, and usually does, undergo, further distillation by different processes, or they may be acid treated, filtered through fullers earth, filter pressed sweated, or sun bleached, each process having its value in producing a product for a certain purpose. For instance, the cut from which general lubricating oils are made is subjected to numerous costly processes before it is ready for the consumer.

Filter-pressing and sweating are certain processes by which the paraffine wax is removed from the stock. Some oils are

sun-bleached or exposed in shallow pans to the sun, which causes them to have a permanent color.

Paraffine-base lubricating oils when finished seldom have a viscosity of over 250 seconds Saybolt, hence, in order to obtain the higher viscosities which are necessary for certain purposes, they must be blended with some other product of higher viscosity. The residue from the Pennsylvania crude is usually selected for this purpose, although various animal or vegetable oils are sometimes used. Asphaltic-base oil can be finished at practically any viscosity and they thus avoid the objectionable necessity of blending.

Paraffine-base oils usually do not have a low cold or pour test, owing to the fact that it is both difficult and costly to remove the paraffine wax which is the cause of their freezing at low temperatures. They will seldom flow at a temperature lower than 30 degrees F. Asphaltic-base oils, not having to contend with the paraffine wax are practically zero cold-test oils, that is, they will still flow at zero. Many of them will not freeze and will still flow at 15 degrees below zero.

This quality is very important when an oil is to be used in or on equipment subjected to extremes of temperatures.

OTHER OILS.

Animal, fish and vegetable oils are sometimes used, either alone or, as usually is the case, as a compounding for mineral oils for different lubricating purposes. The more important are as follows:

Whale Oil.......made from whale "blubber,"
Lard Oil........ made from the fat of hogs,
Neats Foot Oil... made from beeves' hoofs,
Tallow Oil......pressed from beef tallow,
Sperm Oil.......made from the head of the sperm whale,
Cottonseed Oil... made from cotton seed,
Rapeseed Oil.....pressed from rapeseed,

Castor Oil.......made from the castor bean,
Rosin Oil.......made by destructive distillation.

Certain of the above, such as lard and tallow oils, not only have value when compounded with mineral oils, but for some purposes good results can be obtained only by so compounding.

SELECTION OF AN OIL.

It can be safely said that the only way to see what a lubricant is is to actually try it out on the equipment that is to use it; however, it is not always possible to follow this method, and also, even after a certain lubricant has been proven and selected, it is sometimes desirable to have certain specifications as a guide in future purchases. This can only be accomplished through a series of laboratory tests, and the following is a list of those tests generally made. They, of course, need not all be made in selecting oils for certain purposes:

1. Gravity,
2. Viscosity,
3. Color—Odor,
4. Cold Test,
5. Flash,
6. Fire,
7. Carbon,
8. Water and Sediment,
9. Emulsification Tests.

Each of the above tests will be considered in the following with cuts of necessary instruments for making such tests:

Gravity.—By the gravity of an oil is meant its relative weight as compared to that of distilled water. Since a given volume of distilled water at a certain temperature always has the same weight, the weight of the oil to be tested, instead of being expressed in grams or pounds, is said to have a weight which is a certain decimal fraction of the weight of the same

volume of distilled water at 60 degrees Fahrenheit. This decimal fraction is known as the specific gravity of the oil, which may be defined as the ratio of its weight to the weight of an equal volume of distilled water. For example, if a certain volume of oil weighs 21 grams and the same volume of water weighs 25 grams, then the oil weighs 21/25 or 0.840 as much as the water, and accordingly is said to have a specific gravity of 0.840.

Specific gravity can be determined in various ways, one of which is by means of the pyknometer, or specific-gravity bottle, Fig. 2. A special bottle with a perforated stopper is used in the laboratory, but with care an ordinary bottle can be used. The bottle should first be thoroughly cleaned and rinsed and then filled with cold distilled water and heated to 15.6 degrees C., or 60 degrees F. At this temperature the water should just fill the bottle. The full bottle should be carefully weighed, and after emptying and drying it, the empty bottle should be weighed. The bottle should then be filled with the oil to be tested, heated to 60 degrees F., again having the bottle exactly filled with oil and weighed. In each case the weight of the empty bottle must be deducted from the total weight found. The specific gravity of the oil is the ratio between the actual weight of the oil and the actual weight of the water.

To the oil manufacturer or chemist a knowledge of the gravity of an oil is important, and several other instruments are in use to make these determinations, but we have given the simplest means whereby the engineer may make some simple determinations should he so wish. In the first place it is one indication of the crude from which the oil is made. Light-gravity crudes are entirely paraffine-base crudes, and products of such crudes are of correspondingly light gravity. The Pennsylvania, Ohio, Indiana, and northern Louisiana crudes are paraffine-base and light-gravity crudes. The Texas, except from the Electra field in the northern part and the California crudes, are heavy-gravity, and the products made from

8

these crudes are correspondingly heavy in gravity. The experienced oil man can locate with a reasonable degree of accuracy the source from which a certain oil came, and in so doing its gravity is his chief guide. Furthermore, in refining and handling a certain crude, gravity is an important factor, as it is a guide in the distillation process.

It will be remembered that the value of gasoline is determined by its range of distillation and not by its gravity. Of two gasolines having the same range of distillation, the heavier will give the greater power and the greater number of miles per gallon, due to the greater number of heat units. Gravity has no bearing upon the value of kerosene. In the case of lubricating oil, the gravity is of value in that it is an indication of the nature of the crude from which the oil is made. In the case of fuel oil, gravity is of more importance as it is an indication of the heat units in the fuel, the heavier the gravity the greater the number of heat units per gallon.

In selecting a lubricating oil, the oil expert never takes gravity into consideration, as it has nothing to do with the lubricating value. Twenty-five years ago all American oils came from Pennsylvania and the North and were of light gravity. When the heavy-gravity crudes first appeared, though very valuable, they were entirely different; the refiners evidently did not know how to handle them to the best advantage, and for a time the products were the results of practical experimentation on a large scale.

At the present time, however, for many purposes much better lubricating oils are being made out of these heavy-gravity crudes than were even dreamed of five years ago. Properly manufactured oils from the southwestern heavy-gravity crudes have certain natural advantages for some purposes.

Specifications issued by most buyers of oils have now been widened to include oils made from all crudes. For instance, a specification written ten years ago for a gear oil would call

for 25 degrees B. to 26 degrees B. gravity; today, to include the best gear lubricants, the specifications must call for 13 degrees B. to 26 degrees B. which is so wide as to be nearly no specification at all, yet this is exactly the change that is taking place.

Lubricating products are today made from such a wide range of crudes and therefore are of both light and heavy gravity. So far as the consumer is concerned, however, gravity has very little significance, except possibly in the case of fuel oil.

Flash and Fire.—When any liquid is heated sufficiently, it gives off vapor. For example, as water is heated it is converted into steam, gradually until the boiling point is reached. and then rapidly. The same phenomenon occurs in the case of oil, but in this instance the vapors which are given off are inflammable, and form an explosive mixture with the air. The danger of fire or explosion, especially in the case of kerosene or burning oils, led to the introduction of a test which would differentiate those oils which were safe for illuminating purposes from those which were so volatile as to make them dangerous. Later, when the mineral lubricating oils were introduced, the same tests were extended to these heavier oils, not because there was any particular danger of fire in the case of a properly manufactured oil, but rather as a guide to the refiner in his various methods of distillation.

The flash point of an oil is the temperature at which an oil, upon being slowly heated, will give off sufficient vapor to form over its surface a mixture with the air which, upon the introduction of a small flame, will ignite and go out. As the temperature of the oil rises, more vapor is given off, and when the production of vapor is sufficiently rapid, the oil takes fire and burns with a continuous flame. The fire point of an oil may therefore be defined as the temperature at which, upon the application of a flame, the oil will burn continuously.

For the determination of the flash and fire points a number of instruments have been devised, which, with but few exceptions, can be divided into two general classes open cup and closed cup. In the first case the sample is contained in an open reservoir, having free access to the air, and the flash point is the temperature at which sufficient vapor is generated over the immediate surface of the oil to form an inflammable mixture. In the second case the sample is held in a container partially or wholly closed, with, however, sufficient air space to allow the formation of an inflammable mixture of air and vapor. As can readily be seen, the flash point of the same oil will be lower when taken with a closed cup than with an open cup. In order to ignite, it is necessary that there should be a mixture of a certain amount of vapor with a certain amount of air. As the quantity of air is smaller in the closed cup, it will not require the formation of as much vapor to give this mixture; consequently a lower flash point will be recorded when the closed cup is used.

The open cups are best adapted to ordinary work. and, if used with care, give very satisfactory results. Of these the Tagliabue and Cleveland open testers are the most important, cuts of which are herewith shown. There are a large number of other testing cups which space will not permit us to dwell upon.

Flash and fire tests are of special importance in the case of kerosene and petroleum spirits. Burning oils going into different States must be tested with the instruments which these States have legally adopted. While some of the States require no special cups or require only "standard instruments," others specify the Tagliabue, the Foster, or the Elliott instrument. Most of the State laws specifying a minimum flash or fire test for burning oils were drawn up at a time when kerosene was the most valuable constituent of petroleum and when gasoline was a drug on the market. At that time refiners endeavored to increase their yield of kerosense by lowering the

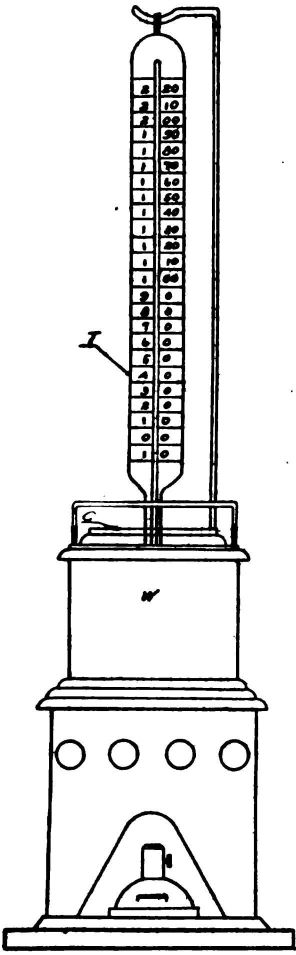


FIG. 3.—TAGLIABUE OPEN TESTER.

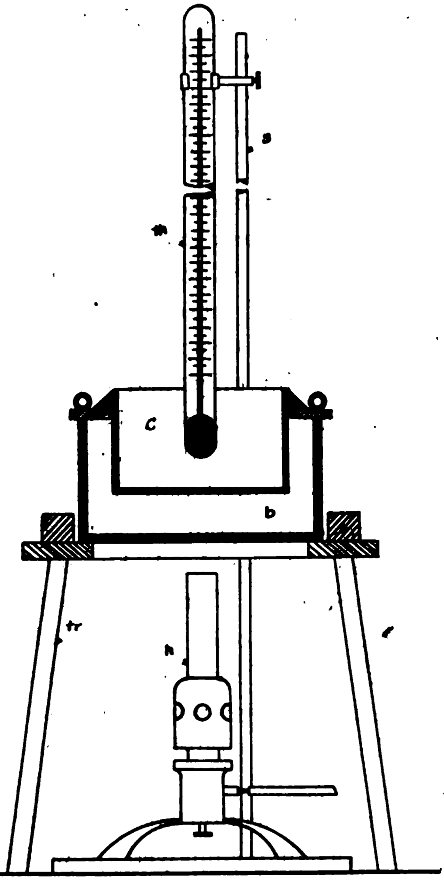


FIG. 4.—CLEVELAND OPEN TESTER.

flash point, with the danger of making a product which had too great a fire hazard, and was dangerous for domestic use. Consequently State laws were devised to prohibit the sale of any burning oils which would be dangerous as shown by the flash point. Today gasoline is more valuable than kerosene, so that these laws are now of less importance.

As far as the flash point of lubricating oils is concerned, it was soon discovered by the refiners of Pennsylvania oils that given lubricating cuts from these crudes would have higher flash points than similar cuts from asphaltic-base crudes. Accordingly, the high flash sales argument for lubricants was used very extensively, and innumerable specifications were drawn up which required a very high minimum flash point. Some of these specifications are still in existence, but they are rapidly disappearing. This sales argument, which is a talking point only, and used by manufacturers who work only on paraffine base oil, is not being pushed as strongly now as at former times, due to the fact that many of the refiners who formerly used only Pennsylvania crude are now compelled to draw from Western and Southwestern crude, and therefore to lower the flash point on their products. For general lubricating oils the only requirement as to flash point is that the flash should be sufficiently high to allow a reasonable margin for safety. As the ordinary bearing temperature would not average more than 150 degrees F., a flash test about 300 degrees F. would leave a margin of more than 150 degrees.

In the lubrication of steam cylinders, when superheated steam was first employed, it was thought necessary to have cylinder oils with a flash point higher than the total temperature of the steam. Superheated steam in some instances reaches a temperature as high as 700 degrees F. It is possible to have a cylinder oil made from certain stocks with a flash point of nearly 620 degrees F., but these oils are so near the end of the distillation that it was found in practice with high

superheat temperatures that the oil that stayed in the cylinder changed very quickly to coke and caused great damage.

The fallacy of the high-flash argument on lubricants for gas-engine oils can be shown by the following facts. The temperature in the explosion chamber of the gas engine cylinder is between two and three thousand degrees Fahrenheit. The maximum temperature of the gases is about 2,700 degrees F., and the average temperature about 950 degrees F. The temperature of the surface of the cylinder walls themselves varies probably from 500 degrees F. at the top travel of the piston rings to 175 degrees F. at the bottom. The piston top and upper rings are undoubtedly over 500 degrees F. When these engines were first built it was thought necessary to use a high-flash oil to withstand this unusual condition. Owing to endless trouble with these heavy oils, however, they have been entirely superseded, and today the lighter oils are almost universally employed. The original idea was to get, if possible, a heavy, high-flash oil that would withstand the high temperature and remain a lubricant. After years of experience it has become the practice to employ a lighter bodied oil that will burn completely and not decompose when it gets into the combustion chamber.

Since there is a vast difference between the temperature of the hot gases and that of the cylinder walls, it follows that the inner and outer surfaces of the oil film will be exposed to quite different conditions. We can consider the oil film as consisting of two layers, the function of one of these being to furnish the lubrication and of the other to withstand the destructive action of the heat and to protect this lubricating layer. The surface exposed to the high temperature of these gases is without doubt affected by this heat, irrespective of its flash point. The outer surface is exposed only to the comparatively low temperatures of the cylinder walls, and, with a film of any appreciable thickness, will be protected from the heat of the burning gases.

Experiments which we have made show that the flash point has nothing to do with the lubricating value of a motor oil. An examination of the flash point of the oil in the crank case, after it had operated in the motor of a car that had run fifty miles, revealed the fact that the flash was only 150 degrees F., while the original flash point was 445 degrees F. The original high flash of the oil did not indicate the sealing qualities necessary for a good motor oil, and with this oil the seal between the piston rings and the cylinder walls was not such as to prevent the gasoline vapor from passing into the crank case during the compression stroke. This gasoline mixture contaminated the oil, made it lighter in body and only 150 degrees F. in flash, thus proving that a high-flash oil would not remain so unless it possessed other and more valuable characteristics, and that an oil having these essential qualifications could well be of a lower initial flash point if its flash would remain constant.

A great many purchasers of steam-cylinder oils, motor oils, and gas-engine oils, as well as many other lubricating oils, think that they require a lubricant having a high flash test. It has been proven in scores of cases that when it comes to practical results that for certain purposes certain low flash point oils are greatly superior to the high flash oils; so much so that among those well informed, the flash and fire points of properly manufactured oils are considered of practically no importance in determining their lubricating value.

Color.—For ordinary purposes the color of an oil can be determined by its appearance in an ordinary sample bottle. Color as well as the general appearance of an oil will vary with the size of the bottle, and it is therefore essential that for comparative purposes the same sized bottle be used. The four-ounce bottle is the standard used by all oil companies, and is therefore the most convenient to use.

In the laboratory, where color must be accurately determined, special instruments are used, the most important being the Lovibond Tintometer, Fig. 5. This consists of a covered

trough or box *T* divided longitudinally by a partition terminating in a vertical knife edge *K* opposite an eyepiece *E* fixed at one end. Two channels *L* are thus formed, which diverge slightly from the end at which the eyepiece is placed. The oil to be examined is placed in a rectangular glass cell *C*, which is placed at the end of the right-hand channel. Stained-glass slides of known depth of color are then inserted into the small

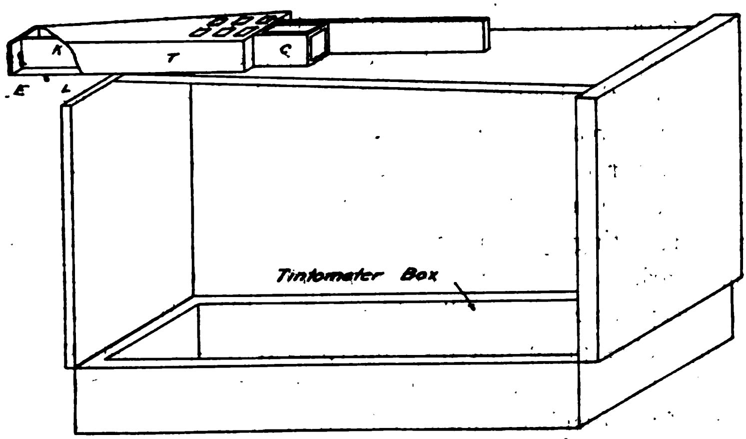


FIG. 5.—LOVIBOND TINTOMETER.

slots at the other end of the channel until the color of the oil is exactly matched. These slides are all numbered on a scale known as the five hundred series, and the depth of color of the oil is expressed numerically. For dark oils a half-inch cell is used, for pale oils a six-inch cell, and for kerosene and gasoline an eighteen-inch cell.

In general, color is no guide as to the suitability of any class of lubricating oil for any particular work. Some pale oils are better lubricants for some purposes than certain red oils, and, on the other hand, there are red oils which for certain purposes are superior to some of the pale oils. A high-grade oil may be either pale or red, and, on the other hand, a lower

grade oil may be either pale or red. The oils of any one refiner can, as a rule, be classed generally by colors, but, owing to the considerable difference in crudes employed and to the manipulation of these crudes, the products of one refiner cannot be compared with those of another on the basis of color alone.

Among the users of oil considerable prejudice has always existed in regard to color. Engineers who are accustomed to dark colored oils regard the light colored oils with distrust. The reason for this feeling is that a thin film of dark colored oil can be seen clearly, and when they use a light colored oil they think the bearings are not properly lubricated because they cannot readily see the film of oil.

The color of any oil changes with continuous use. For instance, if a turbine starts up with an extra pale turbine oil in a circulating system, the color of the oil will gradually become darker, and after running for many months the oil will be a deep wine color. Should a red turbine oil be used, the change takes place just the same, but it is not so noticeable. In Diesel engines, oil also darkens, due largely to unconsumed parts of the fuel being washed down into the crank case. The change in the color does not affect the lubricating properties of the oil provided it is properly filtered by a special process which can be easily and cheaply installed.

When an oil changes in color through use, it does not mean that the oil is becoming worn out or that its lubricating value is being materially affected. It is simply an indication that under the stress of service the oil is taking up various foreign particles which affect the color. Changes in color of an oil, therefore, are due to surrounding conditions and not to the oil itself.

In the case of cylinder oils, also, choice of color in the majority of cases is largely a matter of prejudice. Some engineers prefer a dark cylinder oil, others prefer the bright red filtered cylinder oil. Unless the exhaust steam is to be

used for manufacturing purposes where the oil may get into the articles being manufactured, conditions which can be met by the filtered cylinder oils can be met as well by a dark colored cylinder oil and vice versa.

Odor is a matter of importance both in the case of refined oils and in the case of lubricating oils.

Lubricating oils are classified in regard to odor as normal and compounded. Normal neutral oils are straight mineral oils, and compounded oils are mineral oils combined with fixed oils. The different fixed oils, such as lard, rape and neatsfoot, have characteristic odors which can frequently be recognized in the compounded oil, so that it is usually possible to tell by the odor of an oil whether it is a straight mineral oil, and often, in case it is a compounded oil, what fixed oils have been used in compounding.

All straight mineral oils have a characteristic odor, which, when they are properly manufactured, is almost imperceptible. It occasionally happens that oils which have been stored or shipped in wooden barrels have a rank odor. This is due to the presence of moisture which has affected the glue used in lining the inside of the barrel. Even a small amount of moisture will dissolve the glue and cause the rancid odor. Due to modern facilities, inspection and care, it is practically impossible for moisture to get into the oil at the filling plant. Water, at times, however, does get into the barrels during transportation, and because the importance of keeping the oil barrels dry at all times is not properly understood, the barrels are frequently allowed to stand on end in the open. Snow or water accumulates on the heads, and a small amount will sometimes leak through, causing damage to the oil. To prevent water from getting into the oil in this way, barrels should be stored on the side or bilge; better still, they should never be allowed to stand in the open.

Cold Test.—Nearly all oils are liquid at ordinary temperatures. In the subject of "gravity" reference was made to the

fact that oils expand with an increase in temperature and contract with a decrease in temperature. If the temperature is gradually reduced from 60 degrees F. the oil continues to contract, and finally a point is reached at which it ceases to flow. Oils made from paraffine-base crudes, which contain paraffine wax in solution at ordinary temperatures, cease flowing at higher temperatures than oils made from asphaltic-base crudes. This is due to the fact that wax itself is solid at ordinary temperatures and will, when the oil is cooled, solidify very rapidly so that the oil itself cannot flow.

There are two changes which may occur in an oil when it is cooled, designated as cloud test and pour test. The cloud test is the temperature at which the oil upon being cooled becomes slightly cloudy or partially opaque from the formation of particles of paraffine wax held in suspension by the oil. This test is made only in the case of paraffine-base oils, as asphaltic-base oils contain no paraffine wax and therefore have no cloud test. The pour test is the lowest temperature at which the oil will flow or pour. This test is made on both paraffine and asphaltic-base oils. The term cold test is frequently used loosely to refer to either cloud test or pour test, more frequently to the latter.

There is a fundamental difference between paraffine-base and asphaltic-base oils. In the distillation of a paraffine-base oil, the wax, which is held in suspension by the oil, is a volatile product and is distilled over with the various petroleum distillates, and the residue left in the still after the distillation is complete is petroleum coke. In the case of asphaltic-base oils the asphalt is not volatile and cannot be distilled, and therefore does not distill over with the petroleum products, but remains as a residue in the still. Therefore, while a paraffine-base oil contains paraffine wax in solution, an asphaltic-base lubricating oil does not contain any trace of asphalt.

In making the cloud test, Fig. 6, a cylinder or glass jar about one and one-fourth inches internal diameter and about

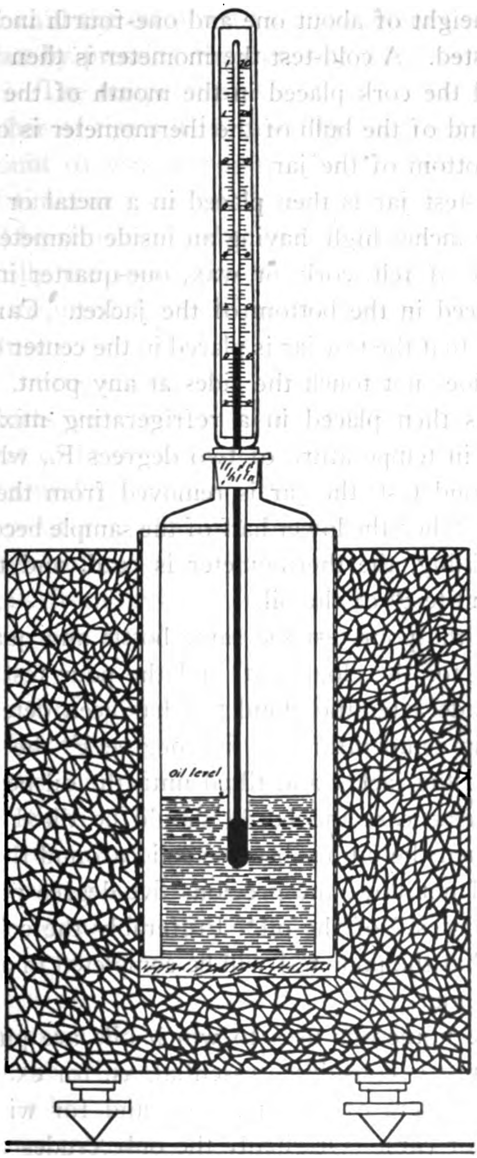


FIG. 6.

five inches long, or a regular four-ounce sample bottle, is filled to a height of about one and one-fourth inches with the oil to be tested. A cold-test thermometer is then inserted into a cork, and the cork placed in the mouth of the jar, so that the lower end of the bulb of the thermometer is one-half inch from the bottom of the jar.

The cold-test jar is then placed in a metal or glass jacket four to five inches high, having an inside diameter of the test jar. A disc of felt, cork, or wax, one-quarter inch in thickness, is placed in the bottom of the jacket. Care should be taken to see that the test jar is placed in the center of the jacket so that it does not touch the sides at any point. The whole apparatus is then placed in a refrigerating mixture and at every drop in temperature of two degrees F., when near the expected cloud test, the jar is removed from the jacket and inspected. When the lower half of the sample becomes opaque through chilling, the thermometer is read, and this is taken as the "cloud test" of the oil.

To make the pour test the same bottle and quantity of oil are used as for the cloud test, and the pour test may, if desired, be made after the cloud test has been determined. At each drop in temperature of five degree F. the bottle is removed from the jacket and tilted until the oil begins to flow. In the extreme case the bottle should be tilted to the horizontal. When the oil has become solid around the thermometer and will not flow, the previous five-degree point is taken as the "pour test" of the oil. Preferably the cold should be applied so that the pour test will be completed in approximately one-half hour.

There are numerous uses to which oils are put for which a low cold test is considered essential, as, for example, in the ammonia cylinders of ice machines, and for winter lubrication of motor cars. Originally the only crudes available for the manufacture of lubricating oils were the Pennsylvania crudes, which have a paraffine base and which contain paraf-

fine wax in solution. To free the oil from this wax it is necessary to chill the oil down to a low temperature and remove the wax by pressing through wax presses designed for this purpose. The temperature to which the oil is congealed and the number of times it is passed through the presses control the amount of wax removed. By repeating the process a number of times at a sufficiently low temperature enough of the wax will be removed so that the oils will have a zero cold test. Naturally the repetition of this process and the lower temperature required for refrigerating increase the expense of manufacture.

With the introduction of oils made from asphaltic-base crudes, products were brought on the market which had a natural cold test of below zero degrees Fahrenheit. These oils possess the natural advantage of being suitable for all extreme climatic temperatures. Russian oils also possess this characteristic of low cold test, but until the manufacture in this country of oils from asphaltic base crudes the amount of low cold test offered to the American trade was practically negligible.

The cold test of cylinder oils is always high. Cylinder oils are usually fed through lubricators that are heated or protected from the cold, and are generally not subjected to cold-weather conditions. No engineer thinks of trying to remove cylinder oil from the barrel until it has been standing in a warm room long enough to thin down. Cylinder oils are not usually made lower than 45 degrees F. cold test and some run as high as 80 degrees F. The high cold-test oils are usually filtered oils and are solid at ordinary temperatures, looking very much like a dark colored vaseline. Some engineers prefer these high cold-test oils, as they think the oils have a heavier body, but the appearance is deceptive for the reason that they contain a considerable amount of low melting-point wax, which makes them solid at ordinary temperatures, but with a slight amount of heat they become thin. All that

is necessary in cylinder oil is to have a sufficiently low cold test to allow it to flow through the lubricator, which may be exposed to drafts from windows or doors in the engine room.

The conditions under which an oil is to be used determine whether or not a low cold test is essential. For general lubrication, where the oil is not exposed to low temperature conditions, a low cold test is unnecessary; on the other hand, where the oil is subjected to such conditions, a low cold test is desirable and, in some cases, necessary.

Viscosity.—In determining the suitability of a lubricating oil for any particular purpose, the most important single characteristic to be considered is its vicosity. In discussing the subjects of color, gravity, flash and fire, and cold test, it has been seen that color and gravity have nothing, and flash and fire practically nothing, to do with the lubricating value of an oil. While the cold test of an oil has an important bearing upon its suitability for feeding uniformly under low temperatures, generally speaking it has nothing to do with the lubricating value, once the oil has reached the bearing surface. Viscosity, however, is a matter of the greatest importance, and should always be taken into consideration, and should, therefore, be thoroughly understood.

The viscosity of an oil is the resistance offered by its particles or molecules in sliding past one another; in other words, its internal or fluid friction. Viscosity denotes the body or relative fluidity of the oil, and is determined by the time it takes a certain amount of oil to flow through a standard orifice. Glycerine and castor oil are very viscous or high in viscosity, and consequently flow slowly, while gasoline and water are low in viscosity and flow more rapidly.

The relative viscosity at atmospheric temperature of two or more oils in sample bottles can be roughly determined by holding the samples together and quickly turning upside down, and then noting the drop from the bottoms of the inverted bottles. The oil which drops first has the lower viscosity,

or the thinner body; if they drop together, the viscosity or body is about the same. Again, the samples may be shaken and the relative viscosity determined by watching the ascent of equal sized air bubbles, the more sluggish the movement the greater the viscosity of the oil.

For the accurate determination of viscosity it is necessary to use an instrument called a viscosimeter. The viscosimeter in general use in this country is the Saybolt Standard Universal. The standard instrument in England is the Redwood, in France the Barbey, in Russia the Lamansky-Nobel, and in Germany the Engler. There is also a number of other instruments in existence, as well as a number of modifications of the instruments already mentioned. Other instruments which have been used to a limited extent in this country are the Tagliabue, the Scott and the Doolittle.

Except for those interested in the export of oil, or those outside of the United States, the Saybolt Universal Viscosimeter is the only instrument with which the oil man needs to be familiar. At the present time there is a decided movement in this country toward its general adoption, so that it has the advantage of being the most generally used and generally known of any of the large number of viscosimeters. It is also simple in construction, is accurately standardized, and has a large bath, by which the oil can easily be kept at the required temperature. The heating arrangements are superior to those of any other viscosimeter on the market.

The Saybolt Universal Viscosimeter, Fig. 7, is made entirely of metal. The cylinder *C* which holds 83 c.c. (cubic centimeters) of oil, is surrounded by a large bath *B*. The upper edge of the cylinder is surrounded by a gallery *G*, into which any surplus oil flows. At the bottom of the cylinder is a jet *J*, through which the oil flows into the receiving flask *F*, which holds 60 c. c. to a mark on its neck. The jet is inclosed by the tube *T*, which extends below the orifice ot the jet, and into the bottom of which a cork *L* with a string attached

9

is lightly inserted, so as to close the tube. The bath is provided with stirrers *S,* fitted with a thermometer for keeping the temperature uniform. The bath is heated by a gas-ring burner *R,* which extends around in a circle under the bath, or by a steam U-tube or by an electric attachment.

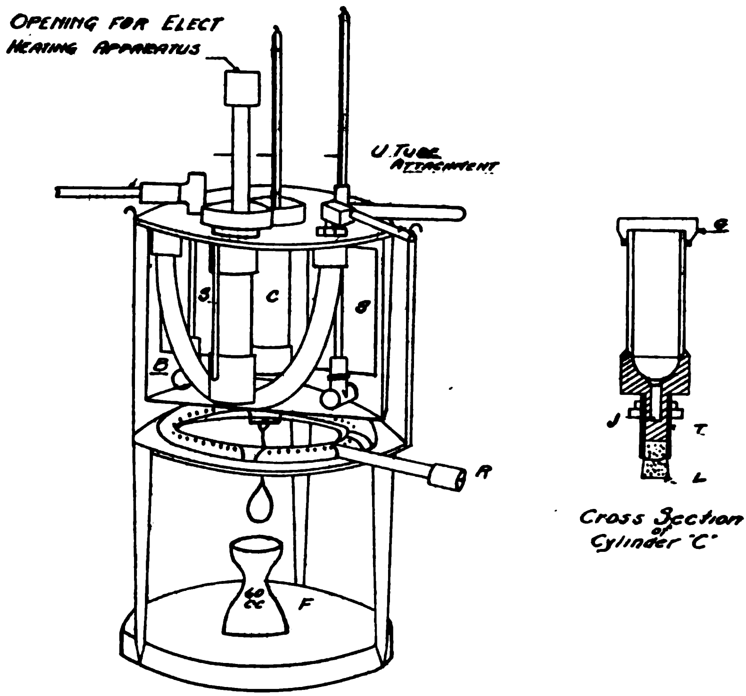


FIG. 7

To make the test the oil cylinder is filled with the oil to be tested, and the bath filled with water or oil which has been brought to the required temperature. The temperature of the oil being tested is adjusted by regulating the temperature of the bath. A thermometer is used to stir the oil in the cylinder, and when this oil is at the correct temperature and

the bath a trifle higher in temperature, the thermometer is withdrawn and the gallery emptied of all surplus oil by means of a pipette. This insures a positive starting point, as the oil will exactly fill the cylinder. The flow of the oil is then started by quickly pulling the string, which withdraws the cork *L* and at the same time a stop watch is started and the time of the outflow of 60 c.c. recorded in seconds. The number of seconds required for the 60 c.c. of oil to pass through the orifice indicates the viscosity of the oil. All Saybolt instruments are so standardized that 60 c.c. of water at 60 degrees F will flow out in 30 seconds, and water accordingly has a viscosity of 30. A light-bodied spindle oil, the time of outflow of which is 75 seconds at 100 degrees F., is said to have a viscosity of 75 at 100 degrees F. A heavy machine oil might have a viscosity of 500 or even higher at 100 degrees F.

Numerous attempts have been made to find the relation between viscosities as determined by the Saybolt Universal, Redwood, Engler, and other viscosimeters. The difficulties are almost insurmountable, however, as the different instruments are standardized in entirely different ways.

The mistake is frequently made of thinking that an oil heavy in gravity is high in viscosity, and that an oil which is light in gravity is of light or low viscosity. The fallacy of this idea becomes apparent when one remembers that water is at the same time thinner in body and heavier in weight than any mineral lubricant. In the case of oils from the same crude, the oil heavier in body would generally be heavier in gravity, but between oils manufactured from different crudes there is no relation between viscosity and weight or gravity.

The temperature at which the viscosity test is made is a matter of the greatest importance, as all oils lose viscosity with a rise in temperature. The temperatures used for the Saybolt Universal are 210 degrees F. for steam-cylinder oils, and 100 degrees F. for engine and machine oils, and as a rule 100

degrees F. for internal-combustion lubricating oils. The temperature of the oil during the test must be kept uniform, as a variation of one degree in temperature would make an appreciable difference in the viscosity. Moreover, in referring to the viscosity of any oil the temperature at which the viscosity was taken should be stated.

In selecting an oil for any particular purpose the viscosity should be determined by the conditions under which the oil is to be used. The purpose of lubrication is to secure and maintain a film of oil sufficient to keep the bearing surfaces apart at working temperatures. If an oil used on a bearing is too light in viscosity at working temperature to sustain the load it will permit metal contact, with a consequent bearing wear and loss of power. On the other hand, if an oil which has too great body at working temperature is used, there will be unnecessary fluid friction, and again there will be loss of power. The more viscous the lubricant, the greater the bearing pressure which can be sustained; the viscosity of the lubricant should, therefore, be in proportion to the bearing pressure. Heavy, slow-moving machinery requires an oil of high viscosity; light, swiftly-moving machinery requires just the opposite. Also, high temperature conditions require an oil of higher viscosity than ordinary conditions. Most of the resistance or lost power in running machinery is caused by the internal friction or the fluid friction of the lubricant used. It is only occasionally, through the use of an improper oil or lack of feed of a proper oil, that actual metal wear on properly constructed bearings takes place.

The general law governing lubrication is that within certain limits a reduction in the viscosity of a lubricant will bring about a reduction in the coefficient of friction, or, in other words, a reduction in frictional heat and in lost power.

Carbon.—While petroleum or any petroleum product is a mixture of hydrocarbons, it must be remembered that the hydrocarbons themselves are chemical compounds. The car-

bon and hydrogen are combined chemically in different proportions to form entirely new and different substances called hydrocarbons. To say that an oil contains carbon means as much as to say that a piece of chalk contains carbon or that water contains hydrogen gas. To be sure, carbon is one of the chemical constituents, but it has absolutely lost its identity, and before the carbon can be reclaimed the compound must be decomposed or broken up into its constituent elements.

When any substance containing carbon is subjected to high temperature without a free supply of air, the substance may be decomposed. If so, the gaseous elements are liberated or combine with other elements to form new compounds, and a residue is left which is chiefly carbon. In most instances, however, this would not be pure carbon, but would consist of various other substances as well. Since this decomposition takes place to a greater or less degree when lubricating oils are subjected to certain conditions, the question of the decomposition of hydrocarbons and the deposit of carbon as a residue is an important one in the lubricating problem.

There is a popular idea that all deposits from a lubricating oil are due to carbonization. This, however, is not the case, as the deposits which are sometimes found in bearings or oiling systems are due to ordinary wear, and consist of small pieces of metal due to metallic wear or iron rust combined with dust and deposits from the surrounding air, and do not contain carbon from the oil unless the bearing has burned out. In the case of steam cylinder lubrication the deposits are due rather to boilerwater impurities and to small amounts of metal, mostly from rust, which are bound together by the oil. In course of time this oil partially evaporates and decomposes, leaving a hard deposit which contains only a small amount of oil and a relatively small percentage of carbon. An analysis of an average deposit from a steam cylinder might show that the residue contained 2 per cent of unchanged oil, 1 per cent of decomposed hydrocarbons, 7 per cent of carbonaceous matter

of carbon, and 90 per cent of ash, consisting of metallic substances, rust, dust, boiler impurities and dirt.

While the deposits found in an internal-combustion engine are by no means all carbon, still the percentage of carbon in the deposit is large, and it is in this case that the question of carbon deposits is of the greatest importance. Any deposit in the cylinder and on the top of the piston and on the valves causes a great amount of inconvenience and damage, and therefore special stress is laid on the amount and nature of the carbon deposited in use for internal-combustion work.

Water and Sediment.—The presence of water in lubricating oil is a matter that is often brought forcefully to the attention of those who handle petroleum products. The same may be said with respect to sediment. Not all of the impurities in petroleum products, however, are included under this heading. Others may be due to improper refining, or to improper or careless handling of the finished products, but their detection usually requires a more exhaustive test or analysis than is possible without the use of a laboratory.

Water in any petroleum product may have come from various sources. It may have been present when the oil was shipped by the manufacturer, it may have entered during transit, or it may have entered at the consumer's plant. It is almost invariably true that refined petroleum products are free from water when they leave the refinery. It is only the rarest exception that proves the rule. But when one considers the handling which the finished product necessarily receives, the piping, storing and transportation, also that it is often necessary, in the case of the heavier products, such as cylinder stocks, to keep the same in tanks where steam coils have been installed, it can be seen readily how easy it is for water to be introduced.

The chances of water being introduced into oil at the consumer's plant are greater than at the plants of the manufacturer. Whenever water is found in lubricating oil at the con-

sumer's plant, a careful examination of the conditions under which the oil is stored or used should be made in order to determine its source. A sample of at least one gallon of the water or mixture of water and oil should be obtained, and if any water from a pipe or faucet could have leaked into the storage tank or could have been in the oil can, a sample of this water should also be taken. The laboratory may then be able to determine whether the water in each case is from the same source. Samples of the oil from the original package should also be secured if possible. Special care should be taken, however, to prevent a sample being taken and placed in a bottle which is not perfectly clean and dry. In obtaining samples from the packages, a glass tube or thief should be used. This should be carefully cleaned and dried before taking samples from the different packages or different grades of oil, and should be cleaned after each sample is taken.

Many complaints concerning the presence of water in lubricating oils have been investigated and in nearly every case the water complained of was of purely local origin, and too much care cannot be taken by those interested in preventing water getting into petroleum products.

The question as to the source of sediment is not as easily answered as that concerning water. Sediment may consist of almost anything. When steel drums are used, a small amount of white sediment, due to the imperfect galvanization of the drums, will sometimes be found. In galvanizing the drums the interior surface is coated with molten zinc. If this process is not perfect the zinc coating will corrode, an action similar to the rusting of iron. Iron gives a red or brown rust, however, while zinc gives a white powder known as zinc white.

In the case of wooden barrels, the usual source of sediment is the glue which is used to coat the inside of the barrel in order to make it oiltight. When this glue has set it is very brittle, and if a heavy sledge is used to start the bung, there is always the possibility that fine particles of the glue lining

will fall into the oil. This is a trouble which the consumer may easily avoid by seeing that barrels of oil of any kind, especially lubricants, are not handled too roughly when being opened. Sometimes the bungs are driven in, and as the bung is made tapering, the outside end, being of larger diameter, often carries in chips and pieces of glue from around the inside of the bung hole. For this reason bungs should not be driven into the barrel, but the stave in which the bung is placed should be lightly tapped until the bung is loosened. Sometimes the barrel head is bored with an auger in order to insert the faucet. If this is not done in a careful manner chips may fall into the oil. In some instances, when oil is pumped from the bung hole, waste is used to wipe off the pump when it is removed, and any oil slopped around the bung hole is also wiped off in this manner. Small particles of waste are thus introduced into the barrel and subsequently a complaint may be made that the oil contained waste. Waste is never used in barrel-filling houses of manufacturers, but is used quite generally in the engine room, and therefore waste found in oil is evidence that the contamination occurred after the oil reached the plant of the consumer. All lubricating oils are carefully strained before being barreled and every precaution is taken to have them perfectly clean when shipped.

In considering water and sediment in lubricating oil it is necessary to distinguish clearly between impurities found in the unused oil and those taken up in use. In circulating or gravity-feed systems supplying oil to either horizontal or vertical engines, a certain amount of water leaks from the piston rods into the circulating oil. This leakage or condensation is, at times, very troublesome, especially where compounded cylinder oils are used, in which event the small amount of fixed oil causes the circulating oil to form an emulsion with the water. Experiments have also proven that the various boiler compounds used to soften the boiler water sometimes come over with the steam or water in priming or

because of foaming boilers. These compounds probably deposit in the cylinders and are afterwards washed out through the stuffing boxes into the circulating oil, having a very bad effect upon the oil, especially if it is mixed with water. Such conditions, of course, make it very difficult for the oil to remain in its original state and do its work properly. This can be remedied, however, by seeing that the stuffing boxes are tight, or by fitting up an arrangement to catch any leaking water so that it will not get into the circulating oil.

In a circulating system water or sediment accumulated in use should be removed by the filters. However, in order that these impurities may be removed from the circulating oil it is necessary to have a circulating system sufficiently large to allow time for the water and sediment to settle out. In many instances the reason for the breakdown of an oil in a system is the fact that it is constantly circulated at high speed with water and dirt, the trouble being due to the size and construction of the system and not to the oil.

Frequently deposits found in bearings are attributed to impurities in the unused oil. An analysis of such a deposit reveals the fact that it is due to impurities accumulated in use and that these impurities are entirely foreign to the oil as received from the manufacturer. These deposits consist of small pieces of metal and dirt which have been collected by the oil in use. The oil acts as a binding or collecting agent for all of the dirt around the plant, for impurities in the boiler water or waste, and the greatest care should be exercised at all times to prevent, to the greatest possible degree, the accumulation of water and sediment in lubricating oils.

Emulsification Tests.—Emulsion tests are made on all straight mineral oils, except cylinder oils. Four emulsion runs are made, usually 40 c.c. of oil in each case and

(1) 40 c.c. of distilled water;
(2) 40 c.c. of salt water;
(3) 40 c.c. of normal caustic soda;
(4) 40 c.c. of boiling distilled water.

The mixture is stirred with a paddle for five minutes at 1,500 revolutions per minute, the mixture being kept at a temperature of 130 degrees F. during the stirring and while settling out. On oils used with forced lubrication or on ice machines the oil must completely settle out in less than 30 minutes.

In concluding this treatise would say that the oil manufacturers are today handling the question of lubrication on the most scientific basis in the way of studying all new devices for the efficient handling of lubricants in operation, and by practical demonstrations this has been accomplished by means of a corps of skilled engineers who are trained carefully in the lubricating business in connection with the laboratory and refinery, using in such demonstrations a number of delicate instruments of all types of mechanism in actual service which proves conclusively the results of their investigations. The system of introducing and selling oils as has prevailed in the past is now becoming obsolete, and the scientific method is being employed. This can have no other result than aiding in the progress of mechanical efficiency, lessening worry and labor of the operating engineer, and giving them subjects to study and improve upon such as they have in general mechanical development where quality is the first consideration.

ENGINES FOR CONVERSION OF ENERGY INTO USEFUL WORK.

By Commander J. O. Fisher, U. S. Navy, Member.

Bacon in his "Novum Organum" states "Man as a minister and interpreter of nature is limited in act and understanding by his observation of the order of nature. Neither his knowledge nor his power extends further." In this statement Bacon recognizes limits between which man must work in understanding the forces of nature and that his understanding is limited by his observations, which, in turn, are limited by his senses.

Man, being a rational animal, reasons and thinks, but his reason and his thought are applied to facts which he observes through his senses.

The engineer must recognize that the acid test for any fact is "Can this fact be established by experiment to our senses?" a demonstration which nature never omits in the trial of the completed machine.

Man observes the universe as being space filled with matter, and since it is matter or the qualities of matter which affect man's senses, the absence of matter cannot be established to his senses (*i. e.,* that while man may assume that space can exist without matter, this is an assumption and not a fact). Man has in the past assumed that a perfect vacuum was possible after extracting all matter from space, only to find that with improved apparatus and by other aids which have been developed, it has later been established that the space still contained matter in minute quantity. Assumptions may be desirable for some particular reason, but when reaching a con-

clusion which involves their use it should never be forgotten that such an assumption was made and affects the conclusion reached.

Exactly as man cannot establish to his senses the existence of space without matter neither can he establish to his senses the existence of matter without space, and since to his senses all space is fixed and invariable he accepts the Law of Conservation of Matter that "the total quantity of matter in the universe is fixed and constant and matter cannot be created or destroyed" as being a rational conclusion based on his observation.

In the study of physical science man has found it necessary and desirable to measure portions of matter by an operation which he calls weighing, and weight is a measure of the relative force with which any portion of matter at any position is attracted by that portion of matter which we call the earth, as compared to the force acting on a specified volume of specified matter at a particular relative position.

The laws having to do with this attraction of one portion of matter for another were discovered and enunciated by Isaac Newton, and are as follows:

1. Every body perseveres in its state of rest or of moving uniformly in a straight line except in so far as it is made to change that state by external forces.

2. Change of motion is proportional to impressed forces and takes place in the direction in which the forces act.

3. Reaction is always equal and opposite to action—that is to say, the action of two bodies upon each other is always equal and in opposite directions.

The law of the conservation of matter and Newton's laws are inseparably connected and it is through the recognition of this conservation law that most new elements of matter have been discovered. That is, if a certain quantity of matter undergoes several changes in structure the quantity of matter remaining should equal the quantity of original matter used

in the operations. After several operations and identification of different portions of matter in their final form as belonging to certain known substances, matter has been found to be missing or present in excess, and in identifying this missing or excess quantity the new elements have been discovered.

A little consideration and thought given to the above shows that while we consider the law of the conservation of matter as established we apply this law by measuring relative forces, and in reality we distinguish between the different elements of matter by their specific gravity, which is a measure of the force of gravity per unit of space occupied by particular matter, and it is rational to regard the universe as made up of different concentrations of force, or, as we say at the present time, of different kinds of matter.

Faraday in discussing this wrote a long dissertion which he delivered before the Philosophical Society called "The Conservation of Force," in which he showed that the conservation of force and the conservation of matter were one and the same (*i. e.,* force could not be separated from matter and matter could not be separated from force), and if the law of the conservation of matter was true it must follow that the law of conservation of force is true.

Faraday also delivered lectures on "The Forces of Nature" in which he demonstrated by experiment how forces, although differing in their characteristics (*i. e.,* effect on our senses), could be converted one into the other. The forces enumerated by him were gravity, gravitation, cohesion, chemical affinity, heat, magnetism and electricity; and the thought arises that the different characteristics of these forces may rest in our means of observing them and is not a real difference (*i. e.,* we observe different forces with different senses, and as they are recognized by their effect on one sense or the other we give them different names).

It is conceivable that there is an ultimate element of matter of which what we call elements at the present time are com-

posed, and the delay in the discovery of this ultimate element is due to the handicap of our senses, and it is also conceivable that the different forces of nature are really combinations or derivations of one ultimate force, which in different combinations is recognized as a different force by our senses.

Prior to the development of the science of chemistry and to the recognition of the law of conservation of matter, new discoveries were haphazard and their establishment on the firm foundation of experimental fact was doubtful. With the development of standards for determining their characteristics and the recognition of the law of conservation of matter new discoveries established on a firm foundation of experimental fact have continued to make their appearance.

Faraday in summing up in his lecture on "The Conservation of Force" states as follows:

"Just as the chemist owes all the perfection of his science to his dependence on the certainty of gravitation applied by the balance, so may the physical philosopher expect to find the greatest security and the utmost aid in the principle of the conservation of force. All that we have that is good and safe, as the steam-engine, the electric telegraph, etc., witness to that principle. It would require a perpetual motion, a fire without heat, heat without a source, action without reaction, cause without effect, or effect without a cause, to displace it from its rank as a law of nature."

Joule in establishing the mechanical equivalent of heat energy, and other discoverers and experimenters who have established units for conversion of energy measurements, have proceeded along the same lines as regards force in combination with matter, and all their discoveries are based upon the Law of the Conservation of Energy, that "the quantity of energy in the universe is constant and energy cannot be created or destroyed."

Therefore since

(*a*) Man cannot establish to his senses the existence of space without matter, or the existence of matter without space.

(*b*) Neither can man establish to his senses the existence of matter without force, or the existence of force without matter and,

In the same manner of observation of nature, man observes action and reaction, cause and effect, motion and position, and in all cases his observations cover an interval of time. Man can assume the interval of time to be infinitely small, but the elimination of that interval is impossible. Therefore we may conclude that—

(*c*) Man cannot establish to his senses the existence of action without time or the existence of time without action.

From (*a*) is derived the LAW OF CONSERVATION OF MATTER or space that "THE QUANTITY OF MATTER OR SPACE IN THE UNIVERSE IS CONSTANT AND IT CANNOT BE CREATED OR DESTROYED."

From (*b*) is derived the LAW OF THE CONSERVATION OF ENERGY (space plus force) that "THE QUANTITY OF ENERGY IN THE UNIVERSE IS CONSTANT AND IT CANNOT BE CREATED OR DESTROYED."

From (*c*) is derived the LAW OF THE CONSERVATION OF POWER (space plus force plus time) that "THE QUANTITY OF POWER IN THE UNIVERSE IS CONSTANT AND IT CANNOT BE CREATED OR DESTROYED."

These CONSERVATION LAWS constitute the basis for all investigations of the forces of nature and their use for the benefit of mankind and on them is erected the entire structure of Engineering.

Therefore to our observation the universe consists of space, force and time, and from the arbitrary limits imposed by our senses, we have derived and accept the CONSERVATION LAWS enumerated above.

To assist in rational discussion definition of terms is necessary and the following are submitted:

Matter may be defined as sensible space.

Force may be defined as a tendency to change sensible space (*i. e.*, matter), its motion, position, volume, structure and force as contained in relative volumes.

Energy is space plus force.

Power is space plus force plus time.

Fundamentally we have only three measurements in the universe: *Force, Space* and *Time.*

In Engineering we have—

Force.

Energy or work (whose factors are force and space).

Power (whose factors are force, space and time).

In the study of Engineering and the conversion of energy into useful work by means of an engine, it is desirable that we consider energy and not force. If a chemist were to consider only one element and neglect the different combinations with the other elements, he would naturally get into error. In the same way an engineer in considering only one of the forces acting to produce certain results and neglecting the others, is sure to get into error.

An instance of this in the study of heat engines and the science of thermodynamics is Carnot's heat engine from which he derives the "Carnot Principle" which was written many years ago.

The following description of Carnot's heat engine and its method of operation is taken from the "Encyclopedia Britannica" and the notes and comment are made in an endeavor to point out the errors in reasoning and fact and the neglect of forces other than heat which make the conclusions reached of doubtful value.

CARNOT'S HEAT ENGINE.

I. Carnot: On the Motive Power of Heat.—A practical and theoretical question of the greatest importance was first answered by Sadi Carnot about this time in his "Reflections

on the Motive Power of Heat" (1824). How much motive power (defined by Carnot as weight lifted through a certain height) can be obtained from heat alone by means of an engine repeating a regular succession or "cycle" of operations continuously? Is the efficiency limited, and, if so, how is it limited? Are other agents preferable to steam for developing motive power from heat?

II. Carnot points out that in order to obtain an answer to this question, it is necessary to consider the essential conditions of the process, apart from the mechanism of the engine and the working substance or agent employed. Work cannot be said to be produced *from heat alone* unless nothing but heat is supplied, and the working substance and all parts of the engine are at the end of the process in precisely the same state as at the beginning.

Note.—To make this paragraph more definite it might be added "And in equilibrium as regards temperature, pressure, etc., and at rest,—(*i. e.,* no unbalanced forces)."

III. Carnot's Axiom.—Carnot here, and throughout his reasoning, makes a fundamental assumption, which he states as follows: "When a body has undergone any changes and after a certain number of transformations is brought back identically to its original state, considered relatively to density, temperature and mode of aggregation, it must contain the same quantity of heat as it contained originally."

Note.—Provided this axiom is correct it must also be true that "when a body has undergone any changes in state, considered relatively to density and mode of aggregation at a constant temperature, it must contain a quantity of heat differing from the quantity of heat it contained originally."

IV. Heat, according to Carnot, in the type of engine we are considering, can evidently be a cause of motive power only by virtue of changes of volume or form produced by alternate heating and cooling. This involves the existence of cold and hot bodies to act as boiler and condenser, or source and sink

of heat, respectively. Wherever there exists a difference of temperature it is possible to have the production of motive power from heat; and conversely, production of motive power, from heat alone, is impossible without difference of temperature. In other words the production of motive power from heat is not merely a question of the consumption of heat, but always requires transference of heat from hot to cold. What, then, are the conditions which enable the difference of temperature to be most advantageously employed in the production of motive power, and how much motive power can be obtained with a given difference of temperature from a given quantity of heat?

Note.—Changes of volume accompany changes in heat content, but all changes of volume are not caused by heating and cooling. In some cases, for instance, the heating and cooling are the result of changes in volume. The necessity for the existence of cold and hot bodies is not clear. If we assume compressed air at a pressure greater than atmosphere and a temperature equal to T, that volume of air will expand (increase in volume) against the pressure of the atmosphere, doing work, and its temperature will fall. If kept in contact with a source of heat at the temperature T its temperature will remain constant, and in this particular case we have a change of volume with no heating and no cooling, and a transfer of heat energy to the working substance which does work by virtue of its original internal energy, utilizing the heat received in maintaining its temperature. This change in volume is accompanied by a corresponding conversion of energy into work, the energy supplied by operating substance being made up by heat from the source of heat and the flow from the source to the operating substance being caused by the difference in temperature. This phenomena is one that can be verified by experiment, and it seems to nullify nearly every statement in the preceding paragraph, being at constant temperature, except that the compressed air during expansion and

while in contact with the source of heat will be at a lower temperature than the source of heat, but this difference in temperature can be assumed as infinitely small.

V. Carnot's Rule for Maximum Effect.—In order to realize the maximum effect, it is necessary that, in the process employed, there should not be any direct interchange of heat between bodies at different temperatures. Direct transference of heat by conduction or radiation between bodies at difference temperatures is equivalent to wasting a difference of temperature which might have been utilized to produce motive power. The working substance must throughout every stage of the process be in equilibrium with itself (*i. e.*, at uniform temperature and pressure) and also with external bodies, such as the boiler and condenser, at such times as it is put in communication with them. In the actual engine there is always some interchange of heat between the steam and the cylinder, and some loss of heat to external bodies. There may also be some difference of temperature between the boiler steam and the cylinder on admission, or between the waste steam and the condenser at release. These differences represent losses of efficiency which may be reduced indefinitely, at least in imagination, by suitable means, and designers had even at that date been very successful in reducing them. All such losses are supposed to be absent in deducing the ideal limit of efficiency, beyond which it would be impossible to go.

Note.—Why is direct transference of heat by conduction or radiation between bodies at different temperatures equivalent to wasting a difference of temperature which might have been utilized to produce motive power? It has not been demonstrated that temperature or difference in temperature produces motive power. Temperature is a measure of relative intensity of heat force and indicates the relative intensity of this force in different matter, and as such indicates the possibility or impossibility of the transfer of heat. In the preceding paragraph the statement is made that production of motive

power from heat alone is impossible without difference in temperature * * * and always requires transference of heat from hot to cold. In this paragraph it is stated that in order to realize the maximum effect there should not be any direct interchange of heat between bodies at different temperatures. This is a most ridiculous form of reasoning. First, the fact is stated (*i. e.*, that a difference in temperature is necessary to transfer heat), and next a goal (*i. e.*, transfer of heat without difference of temperature to prevent waste) is established, which involves the destruction of the previous fact. It is certainly "reductio ad absurdum" to establish a fact that heat cannot be transferred without a difference in temperature and then to indicate the possibility of developing a more efficient heat engine provided this demonstrable fact is eliminated. All of which goes hand in hand with the assumption that temperature can be utilized to produce motive power which under ordinary conception of temperature is as impossible, as it is for velocity or pressure or position to produce motive power. The assumption and statement that a difference of temperature represents a loss of efficiency is the answer to the question we are investigating and trying to establish by reasoning on fact.

VI. Carnot's Description of His Ideal Cycle.—Carnot first gives a rough illustration of an incomplete cycle, using steam much in the same way as it is employed in an ordinary steam engine. After expansion down to condenser pressure the steam is completely condensed to water, and is then returned as cold water to the hot boiler. He points out that the last step does not conform exactly to the condition he laid down, because, although the water is restored to its initial state, there is direct passage of heat from a hot body to a cold body in the last process. He points out that this difficulty might be overcome by supposing the difference of temperature small, and by employing a series of engines, each working through a small range, to cover a finite interval of temperature. Having established the general notions of a perfect cycle, he proceeds to

give a more exact illustration, employing a gas as the working substance. He takes as the basis of his demonstration the well-established experimental fact that a gas is heated by rapid compression and cooled by rapid expansion, and that if compressed or expanded slowly in contact with conducting bodies, the gas will give out heat in compression or absorb heat in expansion while its temperature remains constant. He then goes on to say—(par. VII below)

Note.—In his rough illustration of an incomplete cycle using steam, Carnot fails to show how any reduction in pressure of the steam in the cylinder takes place or how expansion can take place except against a lower pressure in condenser, both questions being neglected. Having a constant pressure on his piston, the expansion of steam at constant temperature against a constant pressure does work at constant pressure and constant temperature. After reaching a point, however, at which all water is steam at atmospheric pressure, further expansion cannot take place unless the pressures on opposite sides of the piston are unbalanced. If the pressure outside is decreased further expansion is possible, but that in itself involves an expenditure of power. If the pressure inside the cylinder (*i. e.*, operating substance) is decreased slightly the pressure on the outside of the piston would operate to compress the operating substance, and this compression can take place at constant temperature and constant pressure, approximately at exactly the same pressure and temperature at which the expansion took place. In this case no useful work is done, and this is what Carnot has established as the general notion of a perfect cycle, admitting that when the steam passes into the condenser, this step having to do with the transfer of heat, does not conform to his requirements as regards the heat cycle. He neglects all mention of pressure which is the operating force generated by the heat change and which, by establishing unbalanced forces acting on the piston, is the indirect means by which the heat transferred to the water, converting it into steam, does work.

In the last statement in this paragraph the fact is omitted that the expansion and contraction of the gas takes place by virtue of the pressure which it exerts or a pressure which is exerted on it, the origin of which is neglected. Heat given out in compression is generated by the work of compression which is obtained from some external store of work, and the heat which is absorbed by the gas in expansion replaces internal energy contained originally by the gas which has been converted into work during expansion. To make an incomplete statement of facts observed in any phenomena and to draw general conclusions therefrom is most dangerous to any investigation of physical truths. For instance, we know that air is supplied to a furnace for the generation of heat, but to draw the inference from this that heat is produced by supplying air to a furnace is so obviously an incomplete statement of fact that few people would make this error. The introduction of air to the furnace is an accompanying circumstance to the introduction of coal and fire, and both are as necessary as the air in generating heat. And we know that a gas will give out heat in compression and absorb heat in expansion while its temperature remains constant, but it can only be compressed by work expended in compressing it and the giving out of heat in compression while its temperature remains constant is caused by the work expended in compressing it. The loss of heat in compression at constant temperature is a result or effect and not a cause. Of course a decrease in volume or compression could be caused by a loss of temperature, which is also loss of heat, but that a gas will give out heat in compression or absorb heat in expansion at constant temperature implies work received from or delivered to some store of work. No transfer of heat at constant temperature during compression or expansion can take place through a transfer of heat only, and work expended or supplied should be mentioned in any statement of facts having to do with this operation.

In paragraph VI Carnot's illustration of an incomplete cycle

uses steam as the operating substance. In the latter part of the same paragraph he refers to a cycle using gas as the working substance. In paragraph VII he uses atmospheric air enclosed in a cylinder as the operating substance. The analogy between the steam engine and Carnot's original statement (see paragraph II) that heat only shall be supplied is not definite, and there is this difference between his statement of conditions in paragraph VII and his illustration of the cycle using steam in paragraph VI. In a steam engine the steam is added at a certain temperature and pressure to the cylinder, the mass of steam is increased during the addition of energy, whereas in the air engine the mass of the air in the cylinder, as per paragraph VII, is constant and the specific heat (*i. e.*, heat per unit mass) increases, otherwise there would be no additional heat. In other words, in a steam engine the mass of the operating substance is increased during the period that heat (*i. e.*, heat energy) is being added, and it is perfectly clear that the "heat" which Carnot refers to in his steam engine and in his air engine, per paragraph VII, are not one and the same. In one case the heat and its carrier, the operating substance, are added to the cylinder contents at the same time. In the other case the heat only is separated from its source and is added to the operating substance.

In the steam engine heat may be considered as energy, as it consists not only of heat but mass. In the air engine heat is a quality of matter, which is subtracted from the source A and added to the operating substance. In this case heat is force, and its transfer is a transfer of force and not energy. In the steam engine the heat is space plus force, in the second case it is force only.

By transfer of mass as in a steam engine work can be done at variable mass, at constant temperature and constant pressure which is at CONSTANT SPECIFIC HEAT (*i. e.*, heat per unit mass), whereas in the air engine work can be done at constant mass, constant temperature and variable pressure which

is at VARIABLE SPECIFIC HEAT (*i. e.*, heat per unit mass). Attention is invited to the fact that in the following illustration of the operation of these engines Carnot uses heat: (1) as being an intangible quality of matter which can be transferred from the source A to the operating substance, and: (2) that this intangible quality of matter is transferred from the source to the operating substance and is the transfer of an intangible quality of matter and not a transfer of operating substance, (*i. e.*, a transfer of force and not energy).

VII. "This preliminary notion being settled, let us imagine an elastic fluid, atmospheric air for example, enclosed in a cylinder *abcd,* Fig. 1, fitted with a movable diaphragm or piston *cd.* Let there also be two bodies A, B, each maintained at a constant temperature, that of A being more elevated than that of B. Let us now suppose the following series of operations to be performed:

Note.—In this paragraph of assumptions regarding the engine no mention is made of the piston being in equilibrium. So far as the statement itself goes the piston can be moving or in equilibrium, or at rest held in position against some force or pressure of the air in the cylinder (*i. e.*, in his description of the operation of the engine the detailed conditions specified are not complete).

"1. Contact of the body A with the air contained in the space *abcd,* or with the bottom of the cylinder, which we will suppose to transmit heat easily. The air is now at the temperature of the body A, and *cd* is the actual position of the piston.

"2. The piston is gradually raised, and takes the position *ef*. The air remains in contact with the body A, and is thereby maintained at a constant temperature during the expansion. The body A furnished the heat necessary to maintain the constancy of temperature."

Note.—What raises the piston? Is it the expansion of the air? and if so, how can the air expand when it is at constant

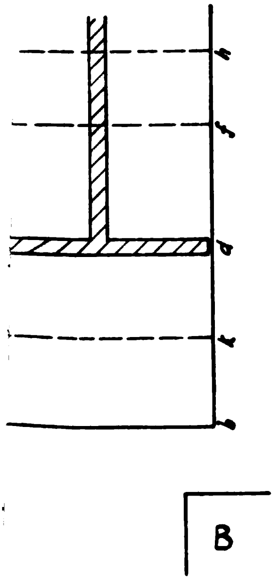


FIGURE 1

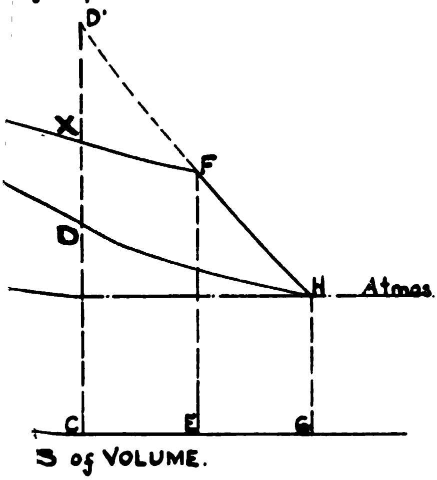


FIGURE 2.

temperature, except by virtue of pressure? That is, the expansion takes place not by virtue of the heat received from the source, but by virtue of the pressure of the air, and as the air expands its temperature decreases and absorbs heat from the source A, and in that way the air is maintained at a constant temperature during the expansion.

"3. The body A is removed, and the air no longer being in contact with any body capable of giving it heat, the piston continues nevertheless to rise, and passes from the position *ef* to *gh*. The air expands without receiving heat and its temperature falls. Let us imagine that it falls until it is just equal to that of the body B. At this moment the piston is stopped and occupies the position *gh*."

Note.—What makes the piston continue to rise except pressure, and the pressure, as we have seen, was not derived from the source of heat but was contained in the air at starting. Until the body A was removed the temperature of the air had not changed, and consequently it could not have had its pressure increased due to an increase in temperature. It could and probably did, however, have its pressure decreased by an increase of volume at constant temperature. The air expands, but it can only expand when the pressure on opposite sides of the piston is unbalanced. However, after expanding to a point at which its temperature is just equal to the body B we can assume that the pressure on opposite sides of the piston is equal.

"4. The air is placed in contact with the body B; it is compressed by the return of the piston, which is brought from the position *gh* to the position *cd*. The air remains meanwhile at a constant temperature, because of its contact with the body B to which it gives up its heat."

Note.—What returns the piston? The air will lose no heat to the body B and the piston will not move except under the action of force, and any heat that is transferred to the body B must come from some source. It can't be created by the air—

it can't be created by the engine—it can only be created by the expenditure of mechanical energy in compressing the air through pressure exerted on the piston. In other words, during this operation the original limitation that heat alone shall be supplied to the engine has been violated unless it is assumed that the work used in compression is derived from the work done in expansion, in which case a liberal interpretation would concede that heat only had been received from an outside source and part of the work done by this heat during expansion had been used during compression in the cycle of operations. To receive heat from an external source at a high temperature to do work and then to use some of that work to deliver heat to a sink at a lower temperature does not appear to me an efficient operation, but rather as progressing backward, in that an engine is used to transfer heat from a source at a high temperature to a sink at a lower temperature when by placing the two in contact it could be transferred much quicker, and more efficiently and without using an engine. What is the particular advantage derived from the operation?

"5. The body B is removed, and the compression of the air is continued. The air being now isolated, rises in temperature. The compression is continued until the air has acquired the temperature of the body A. The piston passes meanwhile from the position *cd* to the position *ik*."

Note.—The compression of the air in this operation involves a further drain on our store of work which we obtained during expansion. This differs from sub-paragraph 4 in that all the work is delivered to the air and is not delivered as heat to some outside sink.

"6. The air is replaced in contact with the body A, and the piston returns from the position *ik* to the position *ef*, the temperature remaining invariable.

"7. The period described under (3) is repeated, then successively the periods (4), (5), (6); (3), (4), (5), (6); (3), (4), (5), (6); and so on.

"During these operations the air enclosed in the cylinder exerts an effort more or less great on the piston. The pressure of the air varies both on account of changes of volume and on account of changes of temperature; but it should be observed that for equal volumes, that is to say, for like positions of the piston, the temperature is higher during the dilation than the compression. Since the pressure is greater during the expansion, the quantity of motive power produced by the dilatation is greater than that consumed by the compression. We shall thus obtain a balance of motive power, which may be employed for any purpose. The air has served as working substance in a heat engine; it has also been employed in the most advantageous manner possible, since no useless re-establishment of the equilibrium of heat has been allowed to occur."

Note.—Statements in this paragraph seem plausible, but the statement that the temperature is higher during the dilation than the compression is one which rests upon the assumption that expansion or dilation at constant temperature through the addition of heat only is a possible physical operation. While as an assumption there are no objections to this, to draw a conclusion that a result is true which depends upon the truth of this assumption is not justified. The same comment applies to the statement "Since the pressure is greater during the expansion, etc." There is no experiment which has ever been made which will establish the fact that the pressure during the expansion of a given volume and mass of gas between certain temperature and volume limits is greater than the pressure during compression of the same volume and mass of gas between the same temperature and volume limits, provided that after expansion and compression the body is brought back identically to its original state, considered relative to density; temperature and mode of aggregation. The statement that "a balance of motive power has been obtained" may or may not be true. It has certainly not been demonstrated. The further statement that no useless re-establishment of the equilibrium

of heat has been allowed to work is not clear. In the process heat has been transferred from a source at high temperature to a sink at a lower temperature. Whether or not this is employing heat "in the most advantageous manner possible" in a heat engine is, to say the least, doubtful.

VIII. "All the operations above described may be executed in the reverse order and direction. Let us imagine that after the sixth period, that is to say, when the piston has reached the position *ef*, we make it return to the position *ik*, and that at the same time we keep the air in contact with the hot body A; the heat furnished by this body during the sixth period will return to its source, that is, to the body A, and everything will be as it was at the end of the fifth period. If now we remove the body A, and if we make the piston move from *ik* to *cd*, the temperature of the air will decrease by just as many degrees as it increased during the fifth period, and will become that of the body B. We can evidently continue in this way a series of operations the exact reverse of those which were previously described, it suffices to place oneself in the same circumstances and to execute for each period a movement of expansion in place of a movement of compression, and vice versa.

IX. "The result of the first series of operations was the production of a certain quantity of motive power, and the transport of heat from the body A to the body B; the result of the reverse operations is the consumption of the motive power produced in the first case, and the return of heat from the body B to the body A, in such sort that these two series of operations annul and neutralize each other."

Note.—That the operations above described can be executed in the reverse order and direction means nothing except that energy cannot be created or destroyed. To state that a process for operating an engine is reversible is merely a statement that energy cannot be created or destroyed. The above paragraph assumes, however, that with the operating substance at the

lower temperature it is possible to return it to the higher temperature. And since the engine receives heat only, we have a plausible statement that heat can be transferred from a cold body to a hot one by a purely self-acting process. The engine may be reversible, but only when it receives work from an external store of work, which violates the original premise in that heat only is received.

X. "The impossibility of producing by the agency of heat alone a quantity of motive power greater than that which we have obtained in our first series of operations is now easy to prove. It is demonstrated by reasoning exactly similar to that which we have already given. The reasoning will have in this case a greater degree of exactitude; the air of which we made use to develop the motive power is brought back at the end of each cycle of operations precisely to its initial state, whereas this was not quite exactly the case for the vapor of water, as we have already remarked."

Note.—The statements in this paragraph are ridiculous. Having performed a series of operations with an engine which violates the original premise and the operation of which is based upon assumptions, it is now stated that it is impossible to produce by the agency of heat alone, a quantity of motive power greater than that which we had obtained in our first series of operations. Having obtained a perfect cycle of operations by certain assumptions, it is impossible to assume anything that will improve the original assumption, and consequently the conclusion reached is that a perfect cycle of operations is performed. This is ridiculous.

XI. "Carnot's Statement of His Principle.—If the above reasoning be admitted, we must conclude with Carnot that the motive power obtainable from heat is independent of the agents employed to realize it. The efficiency is fixed solely by the temperatures of the bodies between which, in the last resort, the transfer of heat is effected. We must understand here that each of the methods of developing motive power attains

the perfection of which it is susceptible. This condition is fulfilled if, according to our rule, there is produced in the body no change of temperature that is not due to change of volume, or in other words, if there is no direct interchange of heat between bodies of sensibly different temperatures."

Note.—Reason can be applied with surety only to fact. To reason upon assumptions is to fall into error. What is the efficiency? Is it the ratio of the net power delivered to the total heat supplied to the engine, or is it the ratio of the net work done to the net heat supplied? These points are not clear. The net heat supplied must be equal to the net work done, otherwise the Law for the Conservation of Energy has been violated. If the efficiency is the ratio of the total heat supplied to the total work, how is the work used in compression, which is negative work, accounted for? If efficiency is equal to the ratio of the net work delivered to the total energy supplied, why is the efficiency fixed solely by the temperatures of the bodies between which is the last resort the transfer of heat is effected? What reasoning on fact has established this principle? That the total quantity of heat which can be utilized by any operating substance in any engine is measured by the temperature limits is not true in fact, except the substance be at constant pressure. Its power of doing work is dependent more upon pressure limits than upon temperature limits and its pressure can co-exist with a low temperature as well as with a high temperature.

XII. "It is characteristic of a state of frictionless mechanical equilibrium that an indefinitely small difference of pressure suffices to upset the equilibrium and reverse the motion. Similarly in thermal equilibrium between bodies at the same temperature, an indefinitely small difference of temperature suffices to reverse the transfer of heat. Carnot's rule is therefore the criterion of the reversibility of a cycle of operations as regards transfer of heat. It is assumed that the ideal engine is mechanically reversible, that there is not, for instance, any

communication between reservoirs of gas or vapor at sensibly different pressures, and that there is no waste of power in friction. If there is equilibrium both mechanical and thermal at every stage of the cycle, the ideal engine will be perfectly reversible. That is to say, all its operations will be exactly reversed as regards transfer of heat and work, when the operations are performed in the reverse order and direction. On this understanding Carnot's principle may be put in a different way, which is often adopted, but is really only the same thing put in different words: The efficiency of a perfectly reversible engine is the maximum possible, and is a function solely of the limits of temperature between which it works. This result depends essentially on the existence of a state of thermal equilibrium defined by equality of temperature, and independent, in the majority of cases, of the state of a body in other respects. In order to apply the principle to the calculation and prediction of results, it is sufficient to determine the manner in which the efficiency depends on the temperature for one particular case, since the efficiency must be the same for all reversible engines."

Note.—There has been no process of reasoning or statement of fact which can be established by experiment which shows that the efficiency of a perfectly reversible engine is greater than the efficiency of any other engine. If a theoretical heat engine is not perfectly reversible the Law of Conservation of Energy is not followed in its operation, and to assume that the Law of the Conservation of Energy is a function solely of the limits of temperature between which an engine works is such a bald misstatement of experimental fact that it is hardly worthy of serious consideration.

To summarize the comment as made in the notes on the Carnot engine and principle as taken from the Encyclopedia Britannica, there are three points in particular which are contrary to the operations of nature and which lead into the manifest errors and conclusions derived from study of the Carnot engine and principle.

The first of these errors is an error in reasoning in that a co-existing phenomena is mistaken for a cause, for instance, expansion at constant temperature doing work. The statement is made that expansion at constant temperature doing work is possible provided the operating substance is in contact with a source of heat. It is not definitely stated that the addition of heat caused the expansion, neither is it definitely stated that the operating substance has an internal energy available as pressure to act on the piston. By omitting these statements of fact the inference is incorrectly drawn that the supply of heat at the same temperature as the operating substance can be used to do work. This is an untenable statement of fact. The expansion of the operating substance under its own internal energy moves the piston, does work, and results in a decrease in the temperature of the operating substance. After this decrease has become an appreciable amount heat flows from the source to the operating substance, maintaining its temperature. The heat does not do the work, it merely maintains the temperature of the operating substance replacing with heat the internal energy used. The work is done by virtue of the internal energy of the air, and if you assume the air at such a condition as regards its internal energy that the pressure on opposite sides of the piston is equal there will be no movement of the piston. This being so, how can these experimental facts be reconciled with Carnot's own statement that heat alone shall be supplied to the engine? Is internal energy present as pressure or heat, or is it energy which can be converted into heat?

The second point is the reverse of the first in that it is assumed that compression of the air can take place as a result of the transfer of heat only at constant temperature. The statement is made that air can be compressed at constant temperature provided it is in contact with a sink of heat at that temperature, the sink of heat receiving heat from the air under compression. It is not stated that the subtraction of heat

causes the compression, neither is it stated that compression takes place as a result of external work being done on the piston which, after causing an appreciable increase in the temperature of the operating substance, will cause a flow of heat to the sink at constant temperature in contact with the operating substance. The fact that there is a flow of heat at approximately constant temperature from the operating substance to the sink is interpreted as being the cause of action. A second time the same error is made and a coexisting phenomena is assumed as the cause of compression. Can this external work done on the piston be considered as heat? Does not Carnot violate his condition that heat alone shall be supplied in receiving external work on the piston?

The third point which is in error is the assumption that the efficiency of a reversible engine is greater than the efficiency of any engine not reversible. It is assumed apparently that there are heat engines which are not reversible, which is equivalent to a statement that the energy supplied to the engine and the energy derived from the engine are not equal (*i. e.*, heat and mechanical energy are not mutually convertible). A reversible engine is nothing more or less than an engine which operates in accordance with the Law of the Conservation of Energy. To assume that any engine can operate except in accordance with that Law is ridiculous, but hardly more so than to assume that because an engine does operate in accordance with that Law its efficiency is unity and a function solely of the limits of temperature between which it works.

When plotted on a pressure volume diagram the indicator card of Carnot's heat engine consists of two isothermal and two adiabatics. Referring to figure 2, lines KF and DH are isothermal lines and the lines FH and KD are adiabatic lines. Contrary to the usual diagram illustrating the Carnot cycle the atmospheric line is also added as indicating the pressure of the operating substance at maximum volume. The cycle can be examined mathematically provided (1) we accept the Law of

Conservation of Energy, and (2) we eliminate the inferences and make our assumptions in accordance with experimental fact. For instance, we can assume that when the piston in figure 1 is at the position *ik* the temperature of the operating substance shall be equal to the temperature of the source of heat A, and at a pressure indicated by the location of a point K on figure 2. We can then go through a complete cycle of operations as follows:

First operation.—Piston at position *ik*, operating substance at Temperature A and at pressure K. Source of heat A is placed in contact with the cylinder. The operating substance expands, driving the piston from the position *ik* to the position *ef* at which point the source of heat A is removed and the cylinder thermally insulated. By the Law of the Conservation of Energy the heat supplied must be equal to the work done by the piston plus the energy change in the operating substance. The work done equals area IKFE (see note), the heat received from A equals QA, the change in the energy content of the operating substance equals QF — QK. We have, in accordance with the Law of Conservation of Energy, our first equation,

$$(1)\ QA = W\overline{IKFE} + QF - QK.$$

Note.—The pressure during expansion of operating substance in driving piston from *ik* to *ef* is indicated by line KF on figure 2. The distance piston has moved is equal to the distance IE on volume line of P. V. diagram, figure 2. Since force times distance equals work or energy the work done is equal to the area IKFE and may be written $W\ \overline{IKFE}$.

Second operation.—The cylinder is now thermally insulated but the pressure of the operating substance on the piston being greater than the pressure of the atmosphere the operating substance will expand doing work and driving the piston from the position *ef* to the position *gh* at which position the pressure of the operating substance is equal to the pressure of the atmosphere and the piston is in equilibrium and at rest. Work done

is equal to area EFHG and energy lost by the operating substance is equal to QF — QH, and since no energy has been received from any other source we have for our second equation in accordance with the Law of the Conservation of Energy:

$$(2)\ QF - QH = W\ \overline{EFHG}.$$

Third operation.—The piston is at position *gh*, the operating substance is at temperature of the sink of heat B, and the pressure of the operating substance is equal to the pressure of the atmosphere. The sink of heat B is placed in contact with the operating substance and work which has been done during expansion is returned to the piston, driving the piston from position *gh* to *cd*, compressing the air, which remains at constant temperature, rejecting heat to the sink of heat B, until its condition is as shown by position D, figure 2, which corresponds to the piston position *cd*. During compression heat has been rejected to B equal to QB. The energy of the operating substance has been changed from QH to QD.

If we assume that during compression of operating substance, by movement of piston from *gh* to *cd*, the sink of heat B was not placed in contact, but that compression was adiabatic (*i. e.*, with operating substance and engine thermally insulated as during the second operation) the compression line would follow the adiabatic line HF to D′ and the work of compression would equal area $\overline{GHD'C}$. Change in energy content QD′ — QH. If then the sink of heat B was brought in contact with operating substance heat would be rejected at constant volume condition until the condition of operating substance would be indicated at D, and we would have heat rejected QB equal to QD′ — QD in accordance with Carnot's Axiom.

In accordance with the Law of Conservation of Energy we have

(3) $W\,\overline{GHD'C} = QD' - QH.$

(4) $QB = QD' - QD.$

(5) $W\overline{GHD'C} - W\overline{GHDC} = W\overline{DD'H}.$

Fourth operation.—The sink of heat B is removed from contact with the operating substance. More work is borrowed from that done during expansion, and the piston is moved from the position *cd*, figure 1, corresponding to D, figure 2, to the position *ik*, figure 1, corresponding to the position K, figure 2, and during this operation the operating substance is thermally insulated and all work expended in compression is added to the operating substance as energy, the operating substance being returned to its original condition as indicated by K. During this operation work has been done equal to area $\overline{CDKI}$. Energy has been added to the operating substance equal to QK — QD, and in accordance with the Law of the Conservation of Energy, we have:

$$(6)\quad QK - QD = W\overline{CDKI}.$$

From these four operations we have derived the following equations:

(1) $QA = W\overline{IKFE} + QF - QK.$

(2) $QF - QH = W\overline{EFHG}.$

(3) $W\overline{GHD'C} = QD' - QH.$

(4) $QB = QD' - QD.$

(5) $W\overline{GHD'C} - W\overline{GHDC} = W\overline{DD'H}.$

(6) $QK - QD = W\overline{CDKI}.$

(7) Net heat received $= QA - QB.$

(8) Net work delivered $= W\overline{IKFE} + W\overline{EFHG} - W\overline{GHD'C} - W\overline{CDKI}.$

By Law of Conservation of Energy we have:

Net heat supplied = Net work done, or,

$$QA - QB = W\overline{IKFE} + W\overline{EFHG} - W\overline{GHD'C} - W\overline{CDKI}.$$

The original question to be answered was "How much work can be obtained from heat alone by means of an engine repeating a regular succession or cycle of operations continuously?"

In order to answer this question it must be demonstrated that the engine has received net heat and has delivered net work, to a reservoir outside of the engine and its operating substance.

Referring to the above equations, it is necessary that for the engine to have received heat that QA — QB must have a finite value, and that the work during expansion stroke must be greater than the work during the compression stroke.

Between condition at minimum volume QK and at maximum volume QH we have the following energy changes in operating substance:

During Expansion.

Received from A = QA.

Received from operating substance QK — QH.

During Compression.

Work, compression from *gh* to *cd* = QD′ — QH.

Work, compression from *cd* to *ik* = QK — QD.

(9) Therefore QA + QK — QH = QD′ — QH + QK — QD.

(10) QA = QD′ — QD.

But, from the third operation we have equation (4)

(4) QB = QD′ — QD.

(11) Subtracting QA — QB = O and QA = QB.

Therefore (1) no ***net heat*** has been received from a source outside of engine and operating substance, and (2) Quantity of heat received equals quantity of heat rejected.

During expansion total work done equals,

$$W\overline{IKFE} + W\overline{EFHG}.$$

During compression total work received equals,

$$\overline{WGHD'C} + \overline{WCDKI}.$$

Net work equals,

$$\overline{WIKFE} + \overline{WEFHG} - \overline{WGHD'C} - \overline{WCDKI} = KXD - XD'F.$$

But, Net work = Net heat.

Therefore QA — QB = KXD — XD'F, which in accordance with equation (11) equals zero.

Therefore

$$KXD - XD'F = O$$

$$KXD = XD'F \text{ and}$$

(1) No net work was delivered by engine.

(2) Work of compression equals work of expansion.

Therefore

1. A heat engine receiving heat only from an external source and delivering net work to an external receiver, and

2. Operating on a Carnot's Cycle consisting of (*a*) isothermal expansion at maximum temperature of cycle.

(*b*) Adiabatic expansion to minimum temperature of cycle,

(*c*) Isothermal compression at minimum temperature of cycle,

(*d*) Adiabatic compression to maximum temperature of cycle and to original condition of operating substance considered relatively to temperature, density and mode of aggregation, and

3. Performing these operations in a regular cycle or succession continually, is an impossible engine and cycle of operations for converting heat only into work.

However the engine has operated and has received heat and converted it into work during expansion of operating substance and during compression the engine has received work and converted it into heat.

But the heat received was at a high temperature and the heat rejected was at a low temperature.

Therefore the net result of a heat engine operating on a Carnot cycle is to receive heat at a high temperature and reject it at a low temperature.

And even this result is based on an assumption that Carnot neglects to mention, viz: that the operating substance has pressure in excess of the pressure on the opposite side of the piston.

If the operating substance at "K" had no excess pressure (*i. e.,* greater than the pressure on the opposite side of the piston) and was at temperature of source of heat A, there would be no expansion or compression, no heat received or rejected, and area KFHD would become a point equal to zero.

NOTES.

THE FRENCH BRIEY-LONGWY IRON ORE BASIN.

We have received a copy of the translation of a secret memorandum which was addressed at the close of 1917 to the German Imperial Chancellor, Count von Hertling, and to Field-Marshal von Hindenburg, by Law Councillor W. Meyer and Dr. J. Reichert, on behalf of the Association of German Iron and Steel Manufacturers, and by General Manager Voegler and Dr. Ing. O. Petersen, on behalf of the German Ironmasters. Dr. Petersen is also the editor of the German periodical "Stahl und Eisen." The memorandum states: (1) That Germany's dependence upon foreign countries for its supply of iron ore constitutes a very grave danger to German industry and to the German State and people. It adds, (2) that forethought for the future renders the displacement of the Lorraine frontier necessary and unavoidable. In the third place, it enlarges upon the point that the mining lands (in France), which it is proposed by Germany to annex, are of enormous value for Germany's national welfare and for her waging of "a future war." A fourth and final chapter is entitled "Conclusion." The whole of the memorandum is very ably analyzed in a Foreword by the translator.

The memorandum states that the iron ore supplies to Germany's enemies are fully assured, whereas the German ironfields will soon be exhausted. Great Britain, it adds, imports a large quantity of ores, and before the war Germany also obtained foreign ores from the same countries which supply Great Britain; but, the memorandum further states, there is a great difference in this respect between the situation of Great Britain and that of Germany, and it arises from the British mastery of the seas. The huge navy of Great Britain ensured for that country, even in time of war, its overseas imports, whereas Germany was at once cut off from almost all the other ore-producing countries. The foreword deals with the comparisons made in the memorandum between the German iron ore resources and those of France, and calls attention to the fact that the German authorities do not mention the Bavarian iron ore deposits, estimations concerning which were given in the German journal "Glückauf" as late as March 12, 1910 ("Die Eisenerzvorkommen in der Fränkischen Alb"), when those deposits were calculated to contain 1,700,000,000 tons, a figure which certainly represents a minimum. The translator adds in this connection that the four German specialists, the authors of the memorandum, exaggerate both the annual consumption of iron ore in Germany and the importance of the French Normandy iron ore deposits, with a view to demonstrate that France, even were she compelled to give up to Germany the Briey-Longwy basin, would still remain in possession of very great iron ore resources. It should be remarked here that the memorandum states most candidly that "few people to this day are aware that Normandy possesses such rich ore deposits, for the German prospectors kept secret from everyone the knowledge they had acquired in the very last years before the war." The Germans were not always so very pessimistic in regard to their mineral resources, for in "Gemeinfassliche Darstellung des Eisenhüttenwesens" (5th edition, 1903, the last edition we have received), a book issued by the Association of German

Ironmasters, the latter, after reviewing the mineral resources of the whole world, congratulate themselves on the favorable position of Germany. The last three lines of the book state the following (translation): " * * * Under all circumstances, the prospects of Germany are more favorable than those of Great Britain, and this applies both to our (the German) coalfields and to the occurrences of ore." Since these lines were written, Germany, it is true, has lost Alsace-Lorraine, but she has ascertained the value of the Bavarian deposits above referred to.

The memorandum throws emphasis repeatedly upon the necessity of shifting the Lorraine frontier farther West owing to the German need of iron ore and to the proximity of many new and immensely important German establishments. It does this after comparing, as we have said, the mineral resources of Germany with those of "Germany's enemies," and then it states, practically in the light of an afterthought, that "by annexing all such necessary frontier areas, there would be incorporated in the German Fatherland a territory which for centuries belonged to the former German Empire." This means that if the Briey-Longwy district were absolutely barren and contained no ironstone in its subsoil, the German ironmasters would not think of annexing it, although it might have belonged formerly to the German Fatherland. Or they might have coveted it simply to increase the distance between France and the German establishments. Germany's "historic right" to the districts in question in this latter case still remains a secondary argument on the part of the German ironmasters who, it would seem, should have based their claim first and foremost upon the standpoint of history. Their brief historical data go as far back as Charlemagne; but the latter is quite as well known in French history as in German history. Charlemagne, in 800, was made Emperor of the West by Pope Leo III, and among the tribes he conquered were the Bavarians and Saxons. By invoking Charlemange, France might possibly lay claim to a part of the present Germany.

The figures quoted for the production of steel in Germany in 1913, the last complete peace year before the war, are as follow, the statements we give in brackets being taken from "Stahl und Eisen": Thomas steel (basic converter steel), 10,630,000 tons, Siemens-Martin steel, 7,977,000 tons. (This is probably made up of open-hearth steel, basic and acid 7,610,000 tons, and steel castings 362,916 tons.) Other steel is given as having reached 328,600 tons (probably made up of crucible steel 99,000 tons, electric steel 77,000 tons, and Bessemer steel 155,000 tons).

The figures for 1916 are also quoted, and are the following:

	Tons.
Thomas steel	7,654,000
Siemens-Martin steel	8,055,000
Other steel	473,300

For both years the figures cover Germany, Alsace-Lorraine and Luxemburg.

The fact that so much basic converter steel should have been manufactured in Germany in 1916, at a time when the country was unable to meet any of its own commercial needs and unable also to export, the whole activities of the nation being exerted in meeting the war requirements, is striking and should not be lost sight of. The output of openhearth steel is practically the same in 1916 as in 1913. The memorandum states that the quantity produced in 1916 would not have reached the figure for 1913, owing to the lack in Germany of resources in discards and scrap, and owing also to the great shortage of labor. "Fortunately," it adds, "the successes of our army enabled us to secure from the conquered territories, besides great quantities of pig-iron, semi-finished and finished products, a large amount of discards and scrap. In this, we were helped, among

other things, *by pulling down the industrial establishments already devastated by war or threatened with destruction."* The italics are ours. We may remind our readers that one of the addressees of the secret memorandum in question was Marshal von Hindenburg, who, we should think, was, better than its authors, conversant with the pulling down activities of the German Army. As it is, the total steel output of Germany was 18,935,600 tons in 1913 and 16,182,000 tons in 1916; the latter figure would hardly point to the alleged "great shortage of labor."

Reverting to the figures for output quoted in the memorandum, it may be surmised that in 1916 no armor plates and very few large-caliber guns, if any, were built for the German Navy; probably no guns, whether for the German Navy or Army were then built of crucible steel. We have always doubted the German assertion that their guns of recent construction were invariably built of that steel, considering the quality now obtainable in the open-hearth furnace. Krupp guns were manufactured before the war at the Cockerill Works, Seraing, and at the Skoda Works, Pilsen. The former works have been completely dismantled, no doubt because they were "threatened with destruction." An official of one of these two works asked whether he built Krupp guns of crucible steel, replied emphatically in the negative; the other firm used occasionally to purchase large naval gun tubes from outside steel makers, who may be said to manufacture tons of open-hearth steel for every ounce of crucible steel they produce. We believe that several Continental works still continue making crucible steel and also puddled iron. They, of course, find use for both metals occasionally, but their main object may be said to be the training of experienced workmen on whom they can rely in an emergency.

We wish to correct an error which has crept in the translation of the memorandum, on page 36. The authors are made to say: " * * * This was more notable from the end of 1870, when German engineers succeeded in putting to practical use the English discovery of the Thomas process * * *" The passage should read: "* * * This was more notable towards the end of the 'seventies" and so forth. The German writers could hardly claim to have improved a process before the discovery of that process. We do not know of any such German improvement of the basic process. In regard to the early statistics of basic and other steel output, they will frequently be found to cover, besides Germany, also Luxemburg and Austria. Even then, in the early 'eighties, the figures of output given in German journals do not show sufficiently high totals to point to any German improvement of the British process in question. The German ironmasters, like the ironmasters in every other country, had to overcome the difficulties inherent to the manufacture of a converter lining, adequate in every respect, before they could make a success of the basic process. The authors cannot have had in mind the "Scheibler process"; this, so far as we can trace, was patented in Germany in about 1886, and it was claimed to require a lesser lime addition, to reduce the duration of the blow and to give a better yield in steel and in phosphoric slag. It was adopted by some German firms and was dropped by several. But, given the British basic process, or, we should say, when a firm had acquired the right against the payment of a royalty to use that process, it would seem to us that that same firm was at liberty to vary at will the addition of baked limestone to suit its own particular requirements and conditions, and this variation would not appear to be patentable, the addition of limestone forming part of the original British invention. Be that as it may, the statement that "German engineers succeeded in putting to practical use the English discovery of the Thomas process" was one which was sure to appeal to von Hertling and von Hindenburg, who probably were not so well versed in steel manufacture as in other subjects. One of our foremost authorities in the manufacture of steel, Sir Robert

Hadfield, stated at the meeting of the Faraday Society on November 12 last, that "German metallurgists * * * have only copied, absorbed and made use of the principles of French, British, American and Swedish metallurgists. I cannot remember any specific instance of the Germans teaching us any basic principles in ferrous metallurgy."

The memorandum also lays great stress upon the very close connection which exists between the steel industry and agriculture, upon the value to agriculture of the phosphates obtained by the basic process. In our present brief review, this point may be viewed simply in the light of a corollary, and we need not refer to it specially.

It mentions "the future war" *ad nauseam* and states that "the displacement of our Lorraine frontiers is absolutely indispensable, not only for the security of the German Empire in a future war, but also for the consolidation of our national welfare * * *" Since Germany not only has not obtained the French Briey-Longwy basin which she has coveted for many years past, but has also lost the portion of that basin which is situated in the provinces she wrested from France in 1871, it would therefore follow that "the future war" now becomes to her an impossibility. But there remain to her the above-mentioned Bavarian deposits of 1,700,000,000 tons, upon which she appears not to have laid much stress hitherto, and the Entente Nations would still have to view contingencies with great caution. The question of "the future war" is outside our province, it is one entirely for the Peace Conference to deal with, but we may be allowed to connect the matter with a statement made many years ago in quite different circumstances by a learned British judge. He said: "The harness of certain horses contains a special strap, called the 'kicking strap.' The horse does not know the strap is there until it wants to kick; and when it wants to kick, it can't." We are quite content to leave all these questions in the hands of the Entente diplomatists who, we feel sure, will do the needful.

In conclusion we recommend the report to our readers. The English version is published in pamphlet form, under the title "German Designs on French Lorraine," at the price of 6*d*. net, by Messrs. George Allen and Unwin, Limited, Ruskin House, 40, Museum-street, W.C. 1.—"Engineering."

IMPROVEMENTS IN THE CONSTRUCTION OF SHIPS.*

The idea of providing a central pipe or duct in the lower portion of a vessel for the purpose of dealing with fluid distributed in different compartments along its length is not new. It is proposed in this particular development to provide the central fore-and-aft duct by dispensing with the ordinary vertical keel plate, and by working instead a relatively narrow box keel running the length of the vessel, and having an uninterrupted passage through it from one end to the other.

Dealing first with the form of this duct keel and its relation to the adjacent ship structure, reference may be made to Figs. 1 and 2.

Fig. 1 represents, in cross section, an ordinary type of vertical keel structure. Fig. 2 represents sections across the proposed duct keel, the framing shown on either side of the keel in Fig. 2 being of the plate-frame type. The maximum width between the vertical plates is not at present intended to exceed 2 feet 6 inches, while it may be as little as 1 foot 9 inches in particular cases. The depth of the duct would generally be the normal full depth of the vertical keel.

*Abstract of paper read before the Institution of Engineers and Shipbuilders in Scotland, by Mr. E. F. Spanner, R. C. N. C.

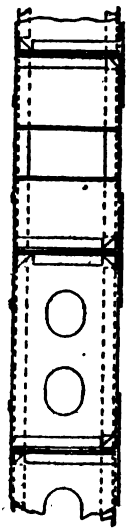


Fig. 1.—Ordinary Form of Vertical Keel.

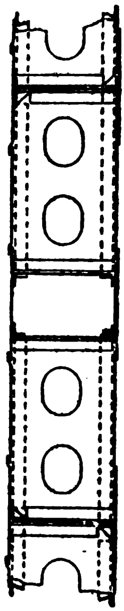
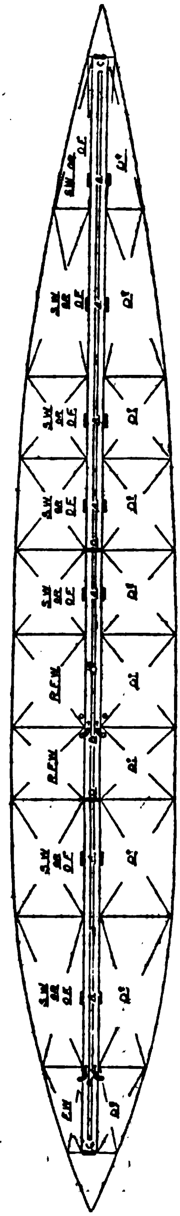


Fig. 2.—Proposed Form of Duct Keel.



A=*Valves in side of Duct.* B=*Pipes through Duct.* C=*Valve for Emergency Trimming.* D=*Divisional Valves.*
E=*Suction for Salt Water or Oil Fuel.*

Fig. 3.—Proposed System of Valves used in conjunction with Duct Keel.

It is at present suggested that in general the poundage of each of the vertical plates forming the duct keel in any vessel should be not less than two-thirds the poundage of the single keel plate proper for that particular vessel, that the poundage of the gutter plate should be 20 per cent in excess of the usual poundage, and that the angles should be not less than 6 inches by 6 inches by ½ inch in section. It is not proposed to increase the thickness of the keel plates, nor is it proposed to fit any stiffening to these keel plates inside the duct. If necessary, stiffening can be fitted across the upper portion of the duct, to support the gutter plate in wake of machinery weights, etc., or for such other purposes as may be desirable.

The actual building of such a duct keel structure as that outlined above calls for few remarks; it is a relatively simple proposition, particularly as this structure would be the first portion of a vessel to be tackled. Sufficient room for riveting is provided, even if the width of the keel is only 1 foot 9 inches, while with a width of 2 feet 6 inches the men would be far from cramped. It would be preferred that the top and bottom angles and the laps and butts of plating should be riveted, and that, after riveting, the edges should be electrically welded instead of being caulked. This would ensure the duct being thoroughly tight and capable of withstanding a good head of fluid. If welding plant is not available, however, a satisfactory job can be secured by proper caulking.

The longitudinal strength of a vessel fitted with a keel structure as proposed is obviously greater than if the ordinary vertical keel is fitted, particularly against hogging stresses.

So far as docking stresses are concerned, if the vessel is properly docked, the members of the keel structure are less severely stressed than is the case with a single vertical keel plate. If she is docked off the center line an unfair stress is brought upon the keel structure, but this is less severe in the case of the box keel than with the ordinary keel. As regards transverse strength, objections may be raised to accepting a box-keel structure without internal transverse-framing, and, as it is desired to dispense with such transverse framing as far as possible, it will not be out of place to deal with the question of transverse strength at some length. In the first place, there is no appreciable reduction in the form-retaining value of the transverse bulkheads, due to the loss of the small area of plating represented by the cross section of the duct keel.

Taking a short transverse section of the vessel, it is generally agreed that the function of the transverse framing is to act as the web of a girder of which the two flanges are the inner and outer bottoms respectively. This girder affords the stiffness necessary, in addition to that provided by the bulkheads, to enable the vessel to maintain her shape. The forces on the bottom of the vessel generally tend to push the bottom into a shape convex from above, *i.e.*, they put the top member of the girder in tension and the bottom member in compression.

It is proposed in the new form of construction to connect the port and starboard portions of the transverse girders to the rigid box forming the duct keel. Constructed in the manner shown in Fig. 2, the duct keel would not collapse except under the effect of stresses which would more than equally injure the structure of a vessel of ordinary construction, and certainly not under any stresses which would normally be transmitted to it through the adjacent structure. If for any particular reason it is desired to fit transverse stiffening to the upper portion of the duct, this is rendered an easy matter by working the upper angles outside the duct.

The sketches and explanation above refer only to the case of a vessel having a double bottom built on the transverse system. The idea is equally easily applied to a vessel designed on the Isherwood system.

It is now proposed to outline the services to which the duct keel can be adapted, considering first the case of a vessel burning coal and using the double-bottom tanks primarily for the carrying of water ballast. In the ordinary way a vessel of fair size would have from 12 to 20 or more double-bottom tanks, and for working them would require a suction leading into each tank. These suctions would each start from a valve box or boxes in the machinery space, and would either run forward and aft above the inner bottom, piercing one watertight bulkhead after another until over their respective tanks, where they would be turned down and led into them through the inner bottom, or else would run through the double-bottom spaces. Such systems necessitate many hundreds of feet of suction pipes and large numbers of holes in the watertight bulkheads or frames and in the inner bottom. Cargo stowage is lost owing to the necessity of protecting these leads of pipes, and constant care and attention is necessary to provide against deterioration. If the vessel is bilged, and damage results to one or more of the suction pipes, special precautions are necessary to prevent water from finding its way through these pipes into compartments far away from the region of the actual damage, and so further jeopardizing the safety of the vessel.

Fig. 3 illustrates the proposed system of dealing with water ballast by means of a duct keel. The duct runs fore and aft between the port and starboard double-bottom tanks, and each tank can be put into connection with it by opening a valve in the side of the duct. These valves are marked A. The duct keel is actually a long double-bottom compartment, and is provided with the usual air escapes. It is proposed that the tail pipe of the ballast pump should have two branches—one leading to the forward end and the other to the after end of this keel compartment. These are shown in the sketch running forward and aft from the suction at E. By using the one or the other, according as to whether the vessel is trimming by the head or by the stern, it would be possible to completely empty any tank or tanks. To fill any tank it is only necessary to open the valve putting this tank into communication with the duct keel and to pump water into the latter until soundings show the tank to be full.

It will be seen that the provision of the duct enables virtually the whole of the ballast-suction system to be dispensed with, while no holes are necessary through watertight bulkheads. Further, water entering the ship following damage cannot find its way along the ship to other undamaged compartments by way of the duct keel, except under the control of the ship's officers. By the adoption of a simple type of pressure-operated valve, control of all the valves on the duct keel can be, if desired, centralized in the engine room, the necessary small pressure leads running along inside the duct.

Returning to the question of damage resulting to a vessel from collision or similar mishap, it will be generally admitted that, of all portions of a ship's structure likely to be damaged at sea, the least likely to suffer is certainly the keel structure. Except grounding, there are probably few cases in which damage has resulted to the vertical keel, and, as the proposed box keel is certainly much stiffer than an ordinary single-plate keel, it is not unreasonable to expect that, even should the vessel take the ground, the duct keel would remain substantially intact. However, to provide against a single damage putting the duct out of operation, two simple-type vertical doors or valves are provided, dividing the duct into three sections. It is not essential that these doors should be more than roughly watertight. They are shown marked D on Fig. 3. If it is

desired to use some of the double-bottom or peak tanks for fresh water or for feed water, the suction pipes to these tanks may be led along from the pump to the tanks by way of the duct keel, thus avoiding the piercing of any watertight bulkheads for the passage of these pipes. This is illustrated in Fig. 3, the pipes being marked B.

Consider now the service which can be obtained from a duct keel in the case of a vessel fitted for burning oil as fuel, and carrying this oil in the double-bottom compartments and in some of the holds. As in the case of the vessel first described, the duct keel could be put into communication with any of the double-bottom compartments containing the oil fuel by the opening of the proper valves, while, by fitting valves in the roof of the duct, it would be possible to put the duct into communication with holds in which fuel oil might be stowed. There are certain points in which the problem of dealing with oil fuel differs from that of handling water ballast, such points arising principally owing to the viscosity of oil at low temperatures. This viscosity is generally so great that it is impossible to pump the oil from the tanks until by some means the temperature of the oil has been raised sufficiently to permit of a considerable lowering of the viscosity. The means usually adopted to heat the oil in the tanks is to fit grids of steam pipes, such grids being necessary, one in each of the tanks used for carrying oil. This is both an expensive and uneconomical procedure, but so far no other means have been found of overcoming the reluctance of the oil to being drawn up through the suction pipes. That the usual steam-heating method is uneconomical is obvious when one considers the fact that, while it is necessary for the oil-fuel suction to be practically at the lowest point of the compartment, the oil itself as it is heated rises to the top, and not until the whole volume in that compartment has been raised in temperature is it generally possible to start a flow which is at all likely to be consistent. Further, only one section of pipe is used at a time, so that 90 per cent or more of the heating system is always out of action and represents useless loss of deadweight. It is further necessary to carefully lag all the suction pipes to prevent stoppage of flow due to cooling of the oil while actually in these pipes, and it is common practice to lead a steam pipe along with the suction pipe, lagging being bound around the two pipes.

It is claimed for the proposed duct-keel system that not only is it possible to eliminate practically the whole of the pipes forming the ordinary filling and suction systems, but also that practically the whole of the heating pipes can be dispensed with. Dealing with the latter claim first, attention is drawn to the following considerations:

1. That the duct keel lies at absolutely the lowest point of the ship, and that, therefore, if easy access is provided from the double-bottom compartments or holds to this portion of the ship, gravity will ensure that flow will result into the duct keel, even if the oil be only semi-fluid.

2. That if steps are taken to heat the oil in the duct keel by means of the usual steam pipes, this oil must necessarily attain a temperature sufficient to enable pumping operations to start much earlier than would be the case were the oil in an ordinary double-bottom compartment, owing to the fact that once the oil has entered the duct it is virtually trapped there, while the volume of oil contained in the duct is comparatively small. Further, as the duct will always contain heated oil when in use, flow from the double-bottom compartments or holds to the duct will be facilitated.

3. It will be quite unnecessary to lag the suction pipes, as these will be extremely short outside the duct, the two tail pipes of the pump leading direct into the duct through the inner bottom.

It will be seen that by fitting heating pipes only to the duct keel, it will be possible to handle the oil effectively, and the very large expense involved in providing for heating pipes to all tanks will be obviated.

As regards the suction arrangements, these will consist of tail pipes from the one or more fuel pumps fitted, and the long leads usually necessary to the several tanks can be dispensed with. Similarly, it will be possible to dispense with the filling pipes to each of the oil-fuel tanks, these tanks being filled by pumping oil into the duct keel and opening the valves to the various tanks in succession.

As has previously been pointed out when discussing the use of the duct keel for water ballast, damage to the duct is unlikely, but if it should occur either forward or aft, the damaged section could be isolated by shutting one or other of the transverse doors. Should some very extensive damage quite put the duct keel out of commission, fuel could still be obtained for a restricted radius of movement from the settling tanks, which would normally contain a certain quantity of oil.

A point worthy of notice is that no loss of double-bottom stowage results from the introduction of the duct keel, as, whether the vessel carries water ballast or oil in the double bottoms, she can, when loaded, carry the duct full of either water or oil, as the case may be. As the duct differs from an ordinary double-bottom compartment only in that it is deeper and freer of access, no difficulty should arise in clearing it of fluid for purposes of inspection, etc.; in fact, it should be a simpler matter to clean out the duct than to clean out an ordinary compartment.

It is now proposed to indicate a further direction in which the new form of keel is likely to be of very considerable value.

The duct keel is of value in the saving of vessels which have suffered serious underwater damage by what is known as the Brunton system. The Brunton scheme contains two main ideas:

1. The addition of water to the damaged vessel so that the maximum corrective effect is obtained from every ton of water added.

2. The utilization of existing powerful pumps or of pumps specially introduced to pump the necessary weight of water rapidly into position. To carry out these ideas it is first necessary to provide that the bulkheads limiting what may be called the emergency-trimming compartments should be watertight and specially stiffened. This is not difficult. It is also necessary to arrange for some connection between the powerful pumps, which would be situated in the machinery compartments, and those emergency-trimming compartments. This Messrs. Brunton have hitherto proposed to do by fitting large pipes running from the pumps to these compartments. The objections to such a proposal are serious—from the naval architect's, builder's and owner's points of view. Such pipes, to obtain the required rate of discharge, must be 12 inches or more in diameter, and they form a source of weakness in the completed vessel, are difficult to arrange for, and are somewhat costly to instal and maintain. It is these several objections that have undoubtedly somewhat hampered the development of this very valuable system.

The manner in which all these objections are met by the utilizing of the duct keel for the purpose of communication between pumps and emergency-trimming compartments hardly needs explaining. The duct keel is in reality a rectangular pipe of very large cross-section, running from one end of the vessel to the other, and to provide communication between it and the forward and after emergency-trimming compartments it is only necessary to fit a suitably operated valve or door at each end. It is an equally simple matter to provide the necessary connection from the powerful pumps to the duct, whether it is decided to utilize the circulating pumps, as has been proposed, or whether other and more flexible pumps are fitted for the purpose.

By the provision of short transverse connections in the double bottom between the duct keel and suitable wing compartments, arrangements could be made to pump water into what might be termed "emergency-keel compartments" for correcting heavy list due to damage, the correction of list being also a feature of the Brunton system.

Reference may be made to the fact that the duct is practically available for use in an emergency, even if it be itself badly damaged. Suppose, for instance, that the fore end is crippled when the forward compartments are bilged. In the first place provision is made for cutting off this portion of the duct from the remainder by a door or valve in the duct, but even if that door or valve is damaged and unable to be closed, it is an undoubted fact that the head over the end of the duct at the undamaged end of the ship is very much less than the head over the damaged end of the duct, so that, by starting the pumps, water is bound to be forced into the emergency-trimming compartment aft, with a beneficial effect on the trim of the vessel.

Another point which may be emphasized is the fact that the duct can be used for the Brunton system, whether its normal service is that of filling and emptying water-ballast tanks or whether it is used for working oil in a vessel fitted for burning oil fuel. Suppose the duct to be full of oil when the vessel is bilged. It is of little consequence whether the fluid pumped into the emergency-trimming compartment is sea water or sea water and oil mixed when the actual safety of the ship is concerned, and there is no reason for varying the procedure to be followed on account of the presence of oil instead of water in the duct.

An effort has been made in the foregoing to touch upon most of the principal features of the form of keel construction recently patented by Mr. J. H. Silley, of the firm of R. & H. Green, and Silley, Weir, Ltd., jointly with the author, but there are a number of points in favor of the idea which could profitably be dealt with at greater length, and others which have not been mentioned.—"Shipbuilding and Shipping Record."

THE CONSTRUCTION OF SHIPS' LINES.

In a valuable paper on "The Design of Lines and Some Considerations of Form," read before the Newcastle Branch of the Association of Engineering and Shipbuilding Draughtsmen, Mr. R. Allan, B.Sc., includes some useful information in regard to methods facilitating the preparation of ships' lines from previous successful designs.

Fig. 1 shows a graphical method of constructing a body plan to give a required displacement when the new vessel is to have the same block and mid-area coefficients as a basis vessel. The method of procedure is first to draw the center, half-breadth, base and load-water lines for the new design on transparent paper or cloth. Then draw in water-lines corresponding to the water-lines of the basis ship at a distance apart given by original spacing multiplied by draught of new ship divided by draught of basis ship. To obtain the offsets for any given water-line, say W_1L_1, place the new body-plan outline over the basis ship body-plan so that points O on corresponding water-lines coincide; then slew upper plan round until the half-breadth spot, W_1, on the new design lies on the half-breadth line, W_1W, of the basis ship, as shown. Project down the various water-line spots of the basis ship as at R, S, B, etc., on the water-line W_1L_1 by lines RR_1, SS_1, BB_1, etc., parallel to W_1L_1 on the new design. The case illustrated is for a new design having a greater breadth

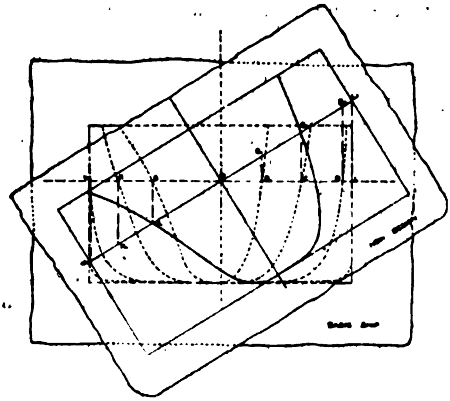


Fig. 1.

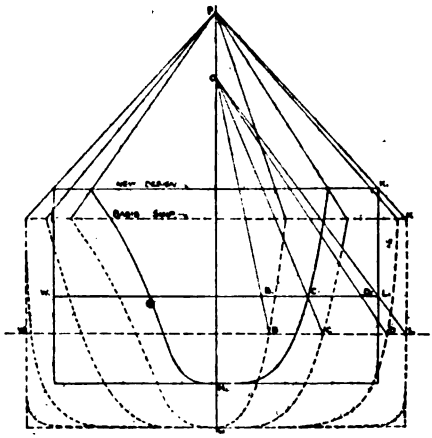


Fig. 2.

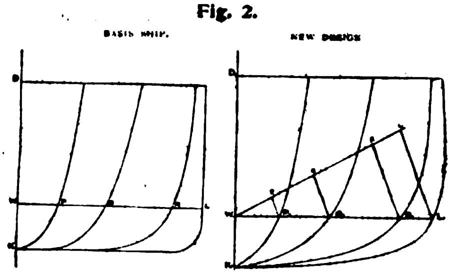


Fig. 3.

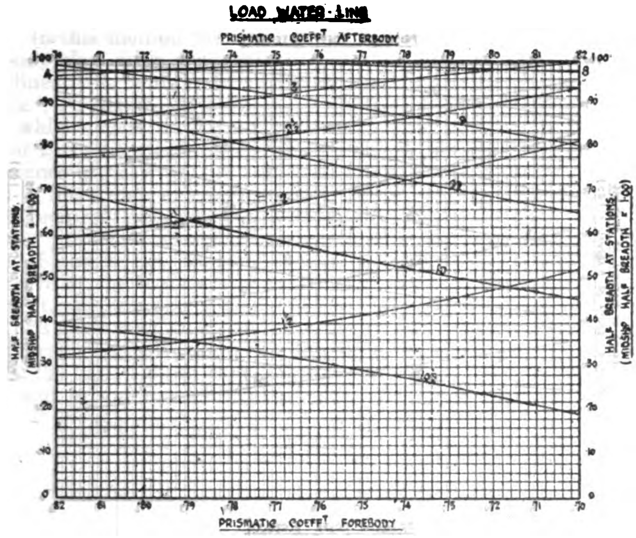

LOAD WATER-LINE
PRISMATIC COEFF^T AFTERBODY
HALF BREADTH AT STATIONS
(MIDSHIP HALF BREADTH = 1·00)
PRISMATIC COEFF^T FOREBODY

Fig. 4.

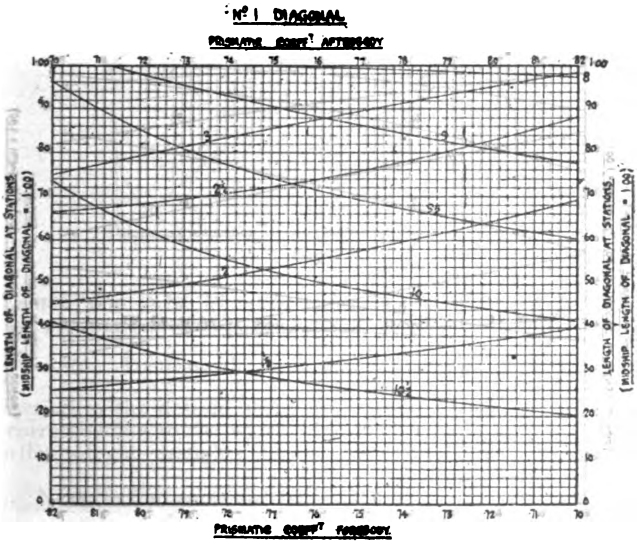

No 1 DIAGONAL
PRISMATIC COEFF^T AFTERBODY
LENGTH OF DIAGONAL AT STATIONS
(MIDSHIP LENGTH OF DIAGONAL = 1·00)
PRISMATIC COEFF^T FOREBODY

Fig. 5.

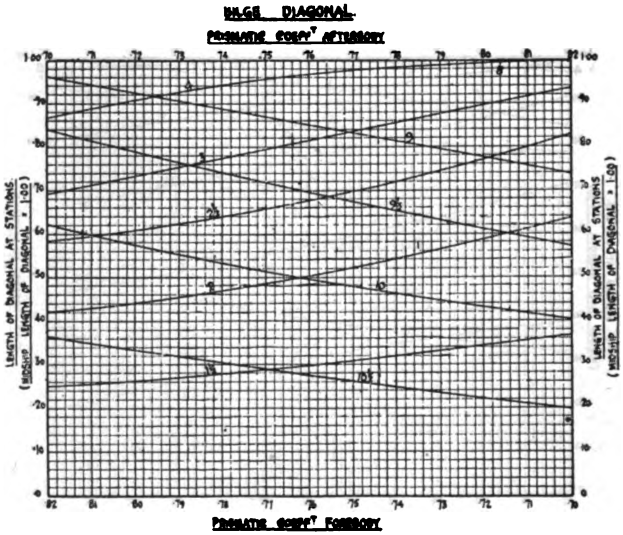

BILGE DIAGONAL.
PRISMATIC COEFF^T AFTERBODY
LENGTH OF DIAGONAL AT STATIONS
(MIDSHIP LENGTH OF DIAGONAL = 1·00)
PRISMATIC COEFF^T FOREBODY

Fig. 6.

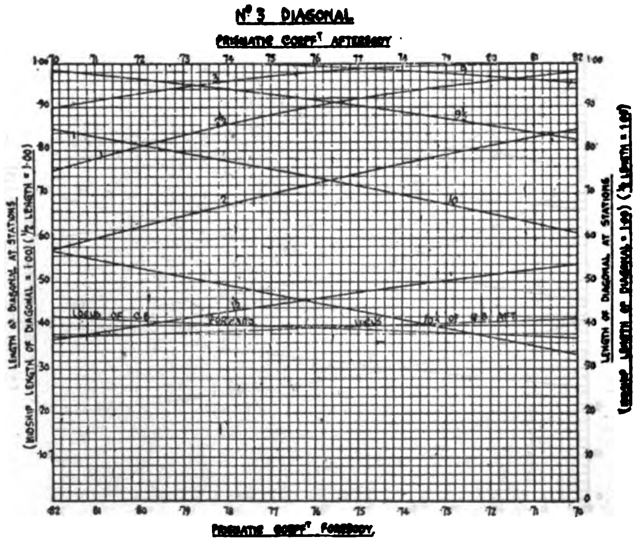

N^o 3 DIAGONAL.
PRISMATIC COEFF^T AFTERBODY
LENGTH OF DIAGONAL AT STATIONS
PRISMATIC COEFF^T FOREBODY.

Fig. 7.

than the basis vessel, but if the reverse be the case the procedure is very similar, the body-plan of the basis ship being in that event placed in the inclined position.

Another graphical method of solving the same problem is shown in Fig 2. In this method the new outline body-plan is placed over the basis ship body-plan with the center-lines coinciding, so that corresponding water-lines WL and W_1L_1 are clear of each other by any convenient distance. To obtain spots on W_1L_1 for new design, join the spots LL_1 by a line which meets the center-line at O. Join OB, OC, OD, etc., to cut W_1L_1 in B, C, D, etc., respectively, which will be the spots required. For convenience in drawing, the angle WLO should be about 45 degrees.

Fig. 3 illustrates a graphical method of constructing a body-plan to give a required displacement, keeping the same form of sectional area curve as the basis vessel. This method is particularly suitable to the design of destroyers, cruisers, channel steamers, etc., where the sectional area curve of the basis vessel has been found a very good one. The displacement and draught being fixed and therefore the block coefficient, the same prismatic coefficient and therefore the same sectional area curve can be kept by simply modifying the mid-area coefficient. As an example, suppose the basis ship to have a block coefficient of .60, mid-area coefficient of .88 and prismatic coefficient of .682, and the block coefficient required

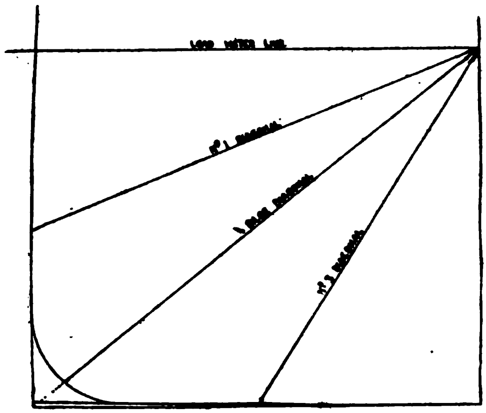


Fig. 8.

for the new ship to be .59. The mid-area coefficient for the new ship to keep the prismatic coefficient the same as for the basis ship must therefore be $\frac{.59}{.682} = .865$. To draw the new ship's body-plan, first design the midship section to the desired shape and coefficient. Then space off waterlines corresponding to the water-lines on the basis ship's body-plan in the proportion of the draughts of the two vessels, *i.e.*, in the proportion $\frac{K_1D_1}{KD}$. Next, to find spots for any water-line W_1L_1 corresponding to WL in the basis ship, alter the offsets of the basis ship in the proportion $\frac{W_1L_1}{WL}$.

This may be quickly performed graphically as shown in Fig. 3. It should be noted that the offsets for any given water-line are altered in the proportion of the corresponding half-breadths of that water-line at the mid-

ship section, and not in the proportion of the half-beams of the ships as was done in the two methods previously described.

When limits of parallel middle body permit, the methods illustrated by Figs. 1 and 2 can by a re-spacing of the sections be extended to cover cases where the block coefficients are different in the vessels under comparison. The procedure in such cases has already been described in "The Shipbuilder."*

Offsets for standardized lines based on the experiments of Baker are given in Figs. 4, 5, 6 and 7. The offsets are for the load water-line and three diagonals placed as shown in Fig. 8. The prismatic coefficients for the fore and aft bodies respectively are selected to give the required longitudinal position for the center of buoyancy. To enable this readily to be done the positions of the centers of buoyancy for each body are included in Fig. 7. The method of procedure is as follows: Suppose, for example, ship 400 feet long, with center of buoyancy 2.6 feet forward, and total prismatic coefficient .750:

$$\frac{2.6}{400} = .0065L \text{ forward, or } .0130\left(\frac{L}{2}\right)$$

Try .765 forward and .735 aft.

Reading from curve of locus of C.B. forward at .765, we find C.B. to be .3975 of ½L forward. Similarly at .735 we have .3785 of ½L aft.

Find difference in moments:

$$\begin{array}{lcl} .765 & \text{at} & .3975 = .3040875 \\ \underline{.735} & \text{at} & .3875 = \underline{.2848125} \\ 1.500 & & \qquad .019275 \end{array}$$

$$\frac{.019275}{1.5} = .01285\left(\frac{L}{2}\right) \text{ forward, which is approximately correct.}$$

An approximate rule is as follows:

$$\frac{\text{C.B. aft or forward} + .1\% \text{ of length}}{\tfrac{1}{2}\text{ length}} =$$

Amount to be added to or deducted from mean C_p. In above example:

$$\frac{2.6 + .40}{200} = .015.$$

C_p forward $= .750 + .015 = .765$
C_p aft $= .750 - .015 = .735$
which in this case is almost identical.—"The Shipbuilder."

INTERNAL-COMBUSTION ENGINE LUBRICATION AND LUBRICANTS.†

By P. H. Conradson.

The main consideration in the selection of oil for the various classes of service are not well understood. Too much weight is given to vague inferences as to the relative value of viscosity and flash point alone, without taking into consideration many other important factors; and while it is true that some few engine manufacturers have conducted exhaustive investigations to determine just what particular grade of oil

---

*See article on "Some Technical Aspects of Shipbuilding Contracts," July number, 1915; No. 59, Vol. XIII.

†From a paper before the American Society for Testing Materials.

should be used by their engines, the greater part of the work of developing suitable lubricants for the various combustion engines has been done by the refiners and dealers of oil.

While the determination of viscosity—that is, body or consistency at a given temperature—is one of the vital tests usually applied to lubricating oil for internal-combustion engines, it is of importance only when the user of the oil is thoroughly familiar with the characteristics of the different crude oils of which it is made and is sure that the mechanical condition calls for the use of an oil having the stated viscosity. Two oils of the same viscosity, made from different crudes, may behave quite differently under identical working conditions.

VISCOSITY TESTS AND TEMPERATURE.

The determination of viscosity of an oil, to be of the greatest value, should be made not only at the lower temperatures, but also as near the temperature of its use as consistent. On the one hand, an oil of low viscosity might be preferable, since it absorbs less power than a thicker oil in the separation of metallic surfaces moving at a high velocity over each other; in other words, high viscosity means a high internal friction. On the other hand, if the oil is too thin it is more easily displaced from between the bearing surfaces, and a heavier oil might then be preferable, as the engine will be more flexible at low speed, owing to better seal and less leakage. Therefore, due consideration must be given to the influence of temperature on the viscosity, and to the necessity of selecting the proper grade for the service required.

Owing to the comparatively high temperature under service conditions, there generally is no trouble from the oil being too thick when once in use. However, an oil must possess a degree of fluidity at ordinary temperatures—which are influenced by climatic conditions—suited to the method of supplying the oil to the working parts, so as to obtain proper lubrication at the start. A lubricating system with exposed pipes, especially a forced-feed system, should have a lower "cold-test" oil to meet these conditions; while in a system where the oil is carried in a pump integral with the engine crank case and where the supply pipes are not exposed, all parts of the system quickly become sufficiently heated to promote a positive circulation and an oil with a higher "cold test" may be used with good results.

The flash point of an oil is of no particular value after the oil has once entered the explosion chamber, where the temperature is considerably higher than that of the flash point. Nevertheless, it is important to use an oil of sufficiently high flash point for reasons of economy if for no other, to resist the vaporization which takes place when the oil comes in contact with the highly heated surface of the piston head, and other parts below the piston. In considering this it is well to remember that with many oils there is little connection between the evaporation (heat) loss and the flash point. Investigation has shown that while two oils may have the same flash points, the percentages of evaporation loss at a given temperature may be quite different. Generally speaking, however, the evaporation loss is greater in oils having a low flash point.

Obviously, that oil which, when subjected to an evaporation test at a certain temperature, loses the least and leaves a residue that has been broken up or altered in character as little as possible by the heat treatment is the most desirable for lubrication of internal-combustion engines. Therefore, the evaporation and oxidation tests offer a promising means of examining the oil and noting the changes that develop in different oils under similar conditions of treatment.

Oil consists principally or entirely of compounds of carbon and hydrogen. At high temperatures these hydrocarbons decompose, forming volatile combustible gases and heavy hydrocarbons, or carbonaceous matter, and free carbon. This breaking up may be due to improper methods of

refining or to the nature of the crude oil. Only the most highly refined and filtered oils should be recommended to meet these conditions.

Much has been written about the carbon residue of the oils for this class of service. The carbon in all oils can be fixed by driving off the oil vapors (without addition of air), leaving a layer of carbon deposit, called carbon residue. The condition of the carbon formation in the cylinder is somewhat different from that found in the carbon-residue test, due to the fact that in the cylinder the oil is spread out in a thin film on hot metal surfaces, and exposed to the burning gases of the explosion, which may carry widely varying amounts of oxygen to combine with the oil. Also, the amount of carbon deposited in the cylinders is governed by the amount of oil reaching the explosion chamber, and may depend to a great extent upon the mechanical fit of the pistons, piston rings and cylinders. It is expected that some oil will find its way into the combustion space, but it is assumed that it will be burned up without depositing an excessive amount of carbon on the cylinder walls and pistons.

One of the most injurious effects of improper lubrication is the formation of this carbon deposit around and under the piston rings. Such a deposit soon renders the spring rings inoperative; they become partly fastened in their grooves, and in this condition form one of the most prolific causes of cut and scored cylinders and broken piston rings. One of the causes of this trouble may be the use of too much oil. From the result of much investigation along this line it has been conclusively demonstrated that an oil with the lowest carbon residue, other things being equal, will leave behind the lowest carbon deposit in the explosion chamber.

In connection with the physical characteristics of the carbon deposit it is of importance to select a lubricant which leaves a loose and flaky or soft carbon deposit, easily removed, rather than one leaving a dense, hard deposit difficult to remove.—"International Marine Engineering."

TOSI DIESEL ENGINES.

We have recently published on several occasions information indicating the rapid extension which is being made in Allied and neutral countries in the internal-combustion engine industry, especially where marine work is concerned, and in this respect the part being played by our Italian Allies is in keeping with their past record of substantial performance in the development of this type of prime mover.

Messrs. Franco Tosi have already built over 400 Diesel engines, these being of both the two-stroke cycle and four-stroke cycle types, totalling, in 1,200 cylinders, about 120,000 B.H.P. The range of horsepower developed hitherto per cylinder is from 15 B.H.P. to 400 B.H.P., and the average power per cylinder being 100 B.H.P., a high average, be it emphasized, considering that the engines comprise besides those used for land power generating stations, pumping sets, marine engines for cargo vessels, submarines, and many for auxiliary service on board ship. The two-cycle engines were built chiefly for the higher powers, *i. e.*, over 200 B.H.P. per cylinder. The speeds of revolution range from 110 r.p.m. to 120 r.p.m. for land engines, to 400 r.p.m. for submarine and high-speed engines. Of marine engines alone Messrs. Franco Tosi have built, or have under construction, 50 engines, totalling 25,000 B.H.P., the largest size hitherto being a six-cylinder unit of 2,400 B.H.P. and the smallest 150 B.H.P.

Messrs. Franco Tosi have decided to concentrate on the four-stroke cycle engine in the future, for they are of opinion that if the same life

is to be obtained from the two-stroke cycle type as from its rival, any advantage of reduced space and weight which it may possess disappears. Advocates of the two-stroke cycle engine give, as one of the principal reasons in favor of this type of engine, the greater power that can be developed in a given size of cylinder, but Messrs. Franco Tosi point out that this advantage is only obtainable at the expense of the greatly increased temperature of the cylinder, piston heads, etc., which arises from the combustion of the larger quantity of fuel necessary for the increased power per cylinder. The ratio of increase in consumption of fuel per cylinder of equal size is, in fact, greater than the ratio of increase of the power obtained from the cylinder, the consumption of fuel per horsepower hour being somewhat greater in the two-cycle than in the four-cycle engine. This greater quantity of heat developed in each cylinder in a given time and the resulting higher temperature is confirmed by the color of the surfaces of the parts exposed to it, and is responsible for the troubles that have been experienced in the two-stroke engine.

It follows, therefore, that the two-stroke cycle engine, if designed with the same size of cylinder, will have a shorter life and may require more frequent overhauling than an engine of the four-cycle type. This is a fact of great importance not only with heavy oil engines for cargo-boat propulsion which must run uninterruptedly at long periods at full load, but also for the lighter type of engine used for submarines, which, although not required to run fully loaded for such long periods, are subjected to high stresses. Considerable progress has doubtless been made with the two-cycle engine, and many improvements introduced into the design, material, cooling of the cylinder and piston, to overcome the difficulties experienced. The modern two-cycle engine is thus undoubtedly more reliable than the older types, but such engines have been in service for too short a time for a definite judgment on them to be arrived at. On the other hand, rapid progress has also been made in the design of the four-stroke cycle engine, whilst many years of highly satisfactory service at full load have already been recorded. Regarded in another way, if an equal life is required from both types of engine—which is what is called for in both land and marine service—an equal quantity of fuel should in both types be consumed in a cylinder of a given size in order to secure equal temperatures. If this condition is adhered to the result will be that the same power will be developed by both types with an approximately equal weight and space.

The crosshead type of construction has been adopted by Messrs. Tosi for all slow-running engines, both for marine and land work, in lieu of the trunk piston design. The chief advantages of the trunk piston engine are simplicity, lightness and compactness. The piston, however, with this type is required to perform the double function—firstly, of maintaining airtightness between the high pressure of compression and combustion on the one side and the crank chamber at approximately atmospheric pressure on the other (*i. e.*, the piston is an obturator); and secondly, of a crosshead in taking the side thrust consequent upon the angularity of the connecting rod. The reliability of its performance in this dual rôle, unless with special attention, is not all that can be desired. When the rings lose their elasticity due to continued exposure to high temperature, or become gummed up, gases leak past the piston ring, and heat up the working surfaces of the piston with the possible result of piston seizure.

The proximity of the gudgeon-pin bearing to the hot piston crown tends, also, to overheat this bearing, giving rise in turn to the danger of distortion and seizing of the main piston. Further, leakage of burnt oil past the piston rings contaminates the crank chamber lubricating oil, causing a more speedy reduction in its lubricating qualities than would arise by normal use. The consumption of lubricating oil, also, is high, due to

the oil which is splashed on to the cylinder walls finding its way into the combustion chamber and so out through the exhaust.

In the present article we propose to deal with the standard Tosi slow-speed four-stroke cycle engine, now being constructed for land and marine work, of various sizes of cylinder units, three, four, six and eight units going to make up the complete engine, according to the power required. For a detailed description reference will be made to Figs. 1 to 4. Of these views, Fig. 1 is a longitudinal combined elevation and section, while Figs. 2 and 3 are two cross-sections, one through a main cylinder and one through the compressor. Fig. 4 is a section, to a larger scale, through a main cylinder head, showing the inlet and exhaust valves.

As already stated, the engine is of the four-cycle type and the six cylinders are approximately 17 inches diameter by 26 inches stroke, developing 83.5 brake horsepower per cylinder, or 500 brake horsepower total in the six cylinders at 180 r.p.m. The piston speed at full speed of revolution is 780 feet per minute, and at full power the corresponding mean effective pressure is 62.5 pounds per square inch. With a mechanical efficiency of 78 to 80 per cent. this gives an indicated mean pressure of about 80 pounds per square inch, which is a moderate figure, allowing a very desirable margin for contingencies and making possible the development of the 500 B.H.P. well within the capacity of the engine at a lesser speed of revolution than the designed figure of 180. We hope to publish at an early date the result of trials at sea of engines of this type. The piston speed is also moderate. The stroke/bore ratio of 1.55 gives ample room for the accommodation of the valves in the cylinder head, keeps the height of the engine within reasonable dimensions (dismantling as will later be explained is largely accomplished from above through the cylinder), and makes possible, if required, the built-up type of construction for the main crankshaft.

The engine is of the slow-speed, enclosed, crosshead, forced-lubrication type, and presents a number of novelties in design. The main framing follows land practice for Diesel engines, and marine practice for steam engines, in that each cylinder with its framing is a separate, interchangeable unit. The jacket, framing, guides and the columns are cast in one piece of grey cast iron of sufficient scantlings to carry the main tension stresses of the piston load. Each unit is bolted up fore and aft, and the whole makes a most rigid structure, although somewhat heavy, since no through-bolts are utilized to take the tension load. The whole framing is enclosed by light metal doors.

Each cylinder and the cylindrical guide faces are machined at the one setting, as is quite common practice with land work, this making for speed of construction and ensuring concentricity. Into each outer cylinder casting a liner of special cast iron is pressed. These liners are of special design, having a spiral rib cast on the outside, in which the bosses for the cylinder lubrication connections are incorporated. The purpose of this spiral rib is to give to the cooling water passing through the jacket a definite path and high velocity, obviating pockets and ensuring equal cooling. The top of this rib is machined and fits into the jacket, so supporting the liner and reducing the thickness required. Each liner is anchored at the top end, and is free to expand at the lower, the water joint at the lower end being made in the usual way.

In regard to the design of the cast-iron framing and bed plate (3), these can be said generally to follow accepted practice. The bed plate (3) is framed of two side I beams, with box-section cross girders carrying the main bearings totally enclosed to form a sump for the lubricating oil, as is usual with forced-lubricating engines. The bed plate is not supported at the center, the cross girders being of ample section to transmit the loads.

To turn to the main running parts of the engine, the piston (4), piston rod, crosshead, connecting rod (5), and crankshaft (2). The piston is quite shallow, since there is no side thrust to be carried and it is fitted with ten Ramsbottom cast iron rings to ensure gastightness. It is connected to a flange on the top end of the hollow piston rod, and is designed for oil cooling. Earlier Tosi engines had water cooling of the main pistons (see "Engineering," July 24, 1912), but oil cooling with this special design of piston is now standardized. In the piston crown is a spiral passage for the cooling oil, which is delivered to the center of the piston crown—the hottest part—through a central pipe in the hollow piston rod, then spirals round the crown and discharges out of the piston crown down the outside of this hollow pipe and so to the crankchamber. It is claimed for this method that the efficiency of cooling is exceptionally high, and so, in spite of the low specific heat of lubricating oil as compared with water, satisfactory results are achieved, and the risk of carbonization of the oil is very much reduced. The objection to water cooling is the danger of leakages of water contaminating the lubricating oil with most disastrous results to the main bearing surfaces. A very small quantity of water mixing with lubricating oil greatly reduces its lubricating efficiency.

On the other hand, lubricating oil is a high priced commodity, and with all Diesel engines the consumption is relatively high, especially in comparison with turbine machinery. Using lubricating oil for the purpose of cooling the pistons tends to lessen the useful life of the oil, and if the supply to the piston crowns should be intermittent carbonization rapidly takes place.

The crosshead adopted on all Tosi engines is of unusual design. With all other designs of Diesel engines the type used has two top end bearings, the gudgeon pin being fixed to the piston rod. With Tosi engines the gudgeon pin is fixed in the connecting rod, and the single bearing is in the piston rod. With this type great care has to be exercised in the design of the connecting rod to ensure a sufficient section of the fork to allow for the bending moment consequent upon the piston load and especially is this the case with a connecting rod drilled for forced lubrication.

The connecting rod of the engine illustrated is relatively short, being 3.8 cranks, and to minimize the side thrust on the guides the center line of the cylinder is set slightly out of line with the center of the crank shaft, as will be seen on reference to Fig. 2. The unique feature of the Tosi cross-head, however, is the gudgeon pin, which, as usual with a single top end bearing, is pressed into the eyes of the forked connecting rod. Outside the gudgeon pin a sleeve is keyed on. The outside of this sleeve forms the bearing surface, and around this sleeve are the usual bearing bushes let into the end of the piston rod. Adjustment for wear is made by the square-headed pins provided with a locking plate, all as shown in Figs. 1 and 2. The gudgeon-pin bearing is force lubricated from the main supply through the drilled connecting rod. The radial slippers of the crosshead are attached to the piston rod and work in the guides—one for ahead and one for astern—formed in the framing as already described. Means for adjustment of the slippers or guides are provided.

With the type of construction adopted the working pressure of the top end bearing is reduced to the moderate figure for forced lubrication of 1,600 pounds per square inch without an unduly long pin being necessary, so minimizing the bending moment for which the forked end of the connecting rod has to be designed. The overall dimensions of the cross-head and slippers are such that, with the bottom end disconnected, the piston rod, crosshead and connecting rod can be withdrawn with suitable tackle through the cylinder. The usual method of dismantling these parts with Diesel engines is to withdraw from below as has often been described in these pages.

Access from below to adjust main bearings, etc., is provided through large crank chamber doors. The chief concern will be the procedure to be adopted in order to clean or renew the main piston rings which require a certain amount of attention, depending upon the quality of fuel oil being used, etc. If a piston can only be withdrawn from the top the cylinder head must be removed, necessitating the breaking of the connections belonging to the cooling-water supply, fuel, fuel injection air, air suction, exhaust, etc. Some of these are high-pressure connections and better left undisturbed. The disconnecting of the valve gear also becomes necessary. On the other hand, removal of the pistons from below often means a higher engine. The connecting-rod bottom is not of the marine type with palm end, but is as shown in Fig. 2, which design tends to reduce its size and so facilitates the consideration of withdrawal through the cylinder.

The crankshaft for this size of engine is in two pieces, a six-throw shaft for the main engine and a two-throw for the compressors, and it is conveniently made of the solid type. It is drilled for forced lubrication. The lubricating oil is delivered by the pump to the main bearings, passing along the crankshaft to the crank pins, up the connecting rod to the gudgeon-pin bearings, through the pipe in the hollow piston rod to the piston, and down outside this pipe to the crank chamber and to the sump in the bedplate from which the lubricating oil pump draws. The pressures at full piston load on the crankpin and main bearings work out at 1,100 pounds per square inch and 540 pounds per square inch respectively.

The cylinder head of cast iron contains five valves, the fuel injection valve (6), two inlet valves (7) and two exhaust valves (8). The provision of two inlet and two exhaust valves, which is exceptional for this size of engine, makes easier the provision of ample passage for the air and exhaust, and the fact of having two small exhaust valves in place of a single larger one renders unnecessary water cooling of the exhaust valves. These valves (see Fig. 4) have cast-iron heads fixed on to steel stems, the better to resist the action of the gases; the baffle to protect the valve-spindle guide from the action of the high-velocity gases will be noted. The arrangement of the valves does not follow normal practice, in that they are not housed in separate cages, but seat direct in the cylinder head. Valve removal requires the lifting of the cylinder head, and a damaged valve seat requires the fitting of a spare head. The casting of the cylinder head is, however, simplified, and the cooling of the exhaust passage by the water circulation is more direct. The valve springs are neatly housed, although inspection of the springs requires dismantling of the valve driving gear.

The inlet valves draw air from the engine room through the grids shown in Fig. 2, through the pipes (11), Fig. 1, from the underside of the working pistons and from the interior of the engine framing (10). The undersides of the pistons are completely isolated by the diaphragm and gland (see Fig. 2) from the interior of the crank chamber. This isolation, which can generally be effected with four-cycle crosshead engines, is most desirable. Gases and burnt lubricating oil mixed with a certain percentage of fuel oil passing the main piston rings would otherwise cause contamination of the lubricating oil in the crank chamber. Leak-offs are provided for this oil as shown, and in certain cases, after filtering, this oil has been used over again as fuel. The suction of the main cylinders and of the compressors being coupled up and taken from this common source, makes for silence, keeps gases from entering the engine room and tends to cool the main engine framing. The quality of air drawn into the main cylinders and compressors is impaired in this way to some extent, and although not of much importance where the main cylinders are concerned, is not a commendable feature with high-pressure compressors (1,000 pounds per square inch delivery pressure).

The nature of the joint between the exhaust manifold (9) and the cylinder heads may be seen from Fig. 2. Allowance is made for expansion, and the supporting feet of the exhaust pipe are free to slide. The exhaust pipe is water-jacketed, with inspection doors opposite each cylinder. The valve gear is of novel design for four-cycle engines, the leading considerations having been, facility of reversing and of disconnecting for removing the cylinder heads.

The camshaft (13) is level with the cylinder tops, and is driven by spiral gearing from the crankshaft. There are two cams for each set of valves, *i. e.,* one set for ahead and one for astern. Motion is transmitted from each cam to a lever, and from the lever mounted on the eccentric fulcrum shaft by means of three pull rods to the adjustable tappets on the bell cranks operating the valves. Driving out one pin in the end of each pull rod frees the cylinder head from the valve gear, and on re-assembling the gear is replaced in exactly the same relation to the crankshaft. The cams run in an oil bath (12), and forced lubrication is provided for the valve gear.

Two tandem compressor cylinders, as will be seen from Fig. 1, are driven from a two-throw crankshaft coupled to the forward end of the main six-throw shaft. These cranks are at right angles to each other. The forward lower cylinder (18) serves to compress air for auxiliary purposes to a pressure of 60 pounds to 70 pounds per square inch, and for starting the engine as later explained. The upper cylinder (17) is the high-pressure cylinder of the three-stage compressor. The lower after tandem cylinder (15) is the low-pressure cylinder of the compressor, and also a starting cylinder, while the upper (16) is the intermediate-pressure cylinder. The capacity of this high-pressure fuel-injection air compressor is such that the swept volume of the low-pressure piston is 18.75 cubic feet per hour per brake horsepower of the main engine. The compressor is provided with automatic valves, as at (22) and (23), to all the stages, and intercoolers reduce the temperature of the air between the stages of compression. Fig. 3.

The design of the driving gear, crossheads and so forth, is exactly the same as for the main engine. At the forward end of this two-throw crankshaft and driven at reduced speed of revolution by spur gearing, are the forced-lubrication oil-circulating pump, cooling-water pump, daily service fuel-oil pump, and bilge pumps suitably grouped together (26). This reduction gearing gives the pumps a speed of revolution of 70 revolutions per minute, and so reduces the shocks of operation of the water pumps.

One of the most interesting features of all marine engines is the arrangement for starting, reversing, control, etc. With the Tosi engine the main cylinders are not used for starting. The disadvantages of introducing starting air to the main cylinders to give the first impulses are considerable. The expansion of the starting air cools the cylinder and its contents, whereas the principle of starting is to revolve the engine at such a speed of revolution that the heat generated by the compression of the air will ignite the injected fuel. For this reason the compression pressure of the Diesel engine is generally governed by the starting condition, and must be such that, allowing for contingencies, sufficient heat will be generated at the relatively low speed of revolution attained when running on starting air to counteract the cooling effect of the expansion of the starting air and ignite the injected fuel. For this reason a much higher compression pressure must normally be carried than, excepting for the starting conditions, would be required. High compressions are not desirable. Again, with slightly abnormal conditions the quantity of fuel that is injected into the working cylinder before ignition takes place may be considerable, and may give rise to high pressures to deal with which safety valves are often fitted to the main cylinders. With the Tosi

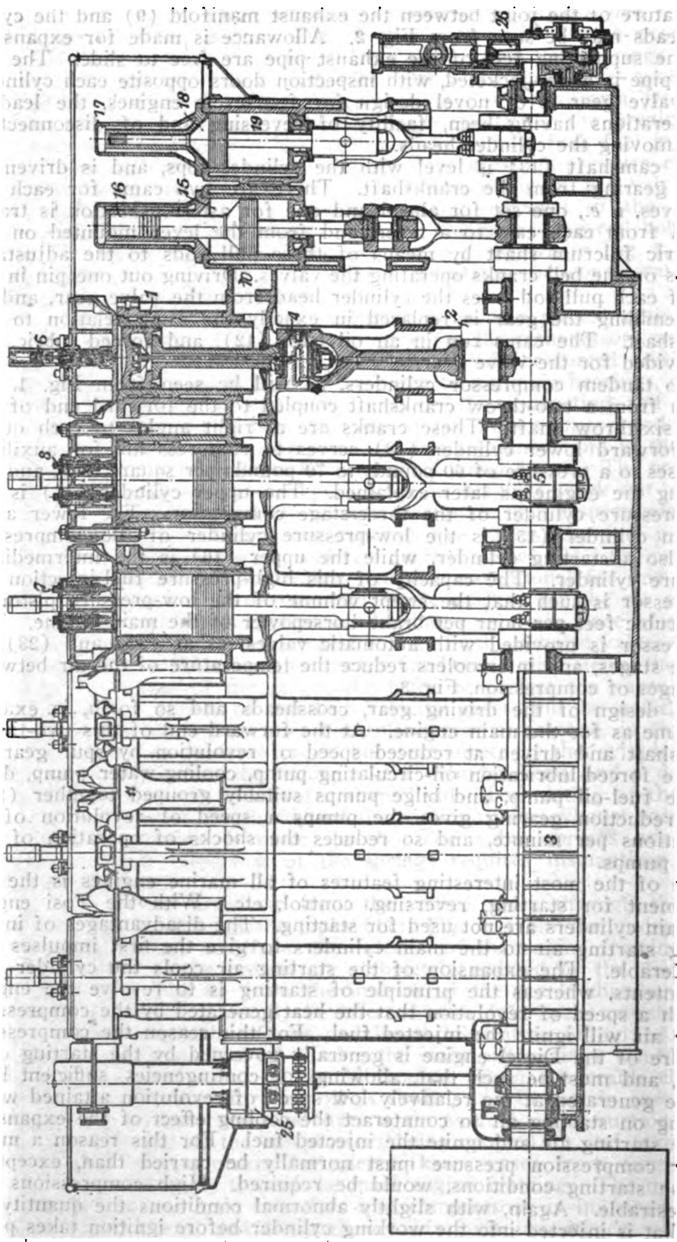


FIG. I.

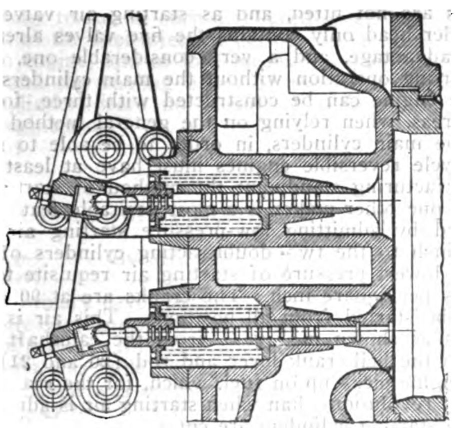


FIG. 4.

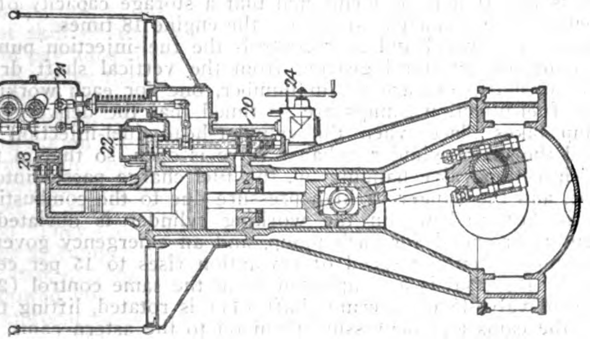


FIG. 3.

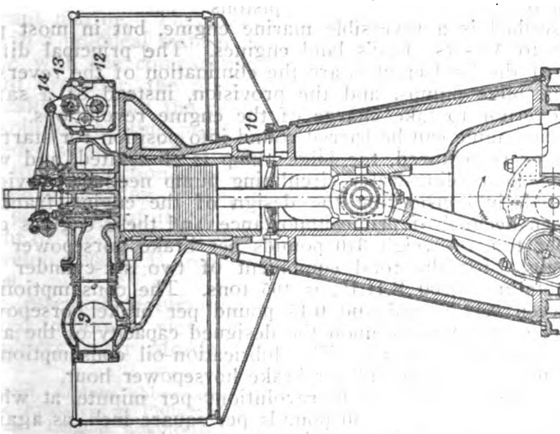


FIG. 2.

engine safety valves are not fitted, and as starting air valves are also eliminated the cylinder head only contains the five valves already enumerated. A further advantage, and a very considerable one, is that by carrying out the starting operation without the main cylinders a marine reversible four-cycle engine can be constructed with three, four, six or eight cylinders, whereas, when relying on the general method employing starting valves in the main cylinders, in order to be able to start from any position, four-cycle reversible engines must have at least six cylinders. From a manufacturing standpoint the method of starting adopted by Tosi and used by one other maker is a most desirable feature.

Starting is effected by admitting low-pressure starting air some 200 pounds per square inch to the two double-acting cylinders of the compressor group. The lowest pressure of starting air requisite to start the engine is 170 pounds per square inch. The cranks are at 90 degrees, so that the engine can be started from all positions. This air is controlled by the piston valves as shown in Fig. 3, from the camshaft driven by special cams through the bell-crank lever and rod (20 and 21).

Immediately the engine picks up on fuel, which, for the reasons already stated, requires fewer revolutions than when starting air is admitted to the main cylinders, these starting cylinders are cut out and function normally as compressors. It may be mentioned that a storage capacity of starting air of 1,000 liters is sufficient to reverse the engine 18 times.

The same hand wheel and gear controls the fuel-injection pumps (25), which are driven by spiral gearing from the vertical shaft driving the overhead camshaft, and are six in number, one for each working cylinder. The fuel-injection pumps are so timed that the delivery stroke of each pump takes place when the cylinder-head fuel-injection valve is closed and the main cylinder exhaust valve is open, so that in the event of a fuel-injection valve hanging up, the fuel charge passes into the exhaust pipe, and the danger of high pressure due to the combustion of an accumulation of fuel within the working cylinder is obviated. Hand adjustment is provided for each pump, and an emergency governor cuts out the pumps when the speed of revolution rises to 15 per cent above the normal. Reversal is accomplished from the same control (24). The cylinder-head valve-lever fulcrum shaft (14) is rotated, lifting the ahead levers off the cams and depressing them on to the astern cams, and also altering the setting as required of the piston valves controlling the entrance of the low-pressure air to the two starting cylinders. Sight-feed forced lubrication is provided to the main pistons.

The engine described is a reversible marine engine, but in most particulars is similar to Messrs. Tosi's land engines. The principal differences in the case of the land engines are the elimination of the reversing gear, double cams, bilge pumps, and the provision, instead of a safety governor, of a governor to take charge of the engine revolutions. For land work, since the engine can be barred round into position for starting, only one compressor is required, the bilge pump is eliminated, and with certain installations no cooling-water circulating pump need be provided.

As already stated, in considering the design of the engine framing, weight has not been deemed of prime importance, and these engines give some 7 B.H.P. per ton, or weigh 310 pounds per brake horsepower developed. The weight for the total equipment of two six-cylinder 500 B.H.P. engines, totalling 1,000 B.H.P., is 165 tons. The consumption of fuel oil lies between 0.42 pound and 0.45 pound per brake horsepower hour, depending in some measure upon the designed capacity of the auxiliaries driven by the main engine. The lubrication-oil consumption is between 0.007 pound and 0.01 pound per brake horsepower hour.

The engine will fire regularly at 40 revolutions per minute, at which speed the compression pressure is 450 pounds per square inch, as against 485 pounds per square inch at 165 revolutions per minute. One reason

for the capacity to run at such a reduced speed is the fact that the phase and duration of the fuel-injection period can be regulated according to the speed of the engine, and at low speeds the quantity of fuel-injection air is reduced and the cooling effect of the excessive quantity that would enter the cylinder, were no means of regulation provided, is entirely absent.—" Engineering."

PROGRESS IN TURBINE SHIP PROPULSION.*

By Mr. FRANCIS HODGKINSON.

The extraordinary development of the steam turbine for land purposes, almost entirely supplanting the reciprocating engine, rendered the application of steam turbines for the propulsion of ships a natural sequence.

The *Turbinia* was built especially to demonstrate the possibility of turbine propulsion and was of torpedo-boat type, 100 feet long, 9 feet beam, 3 feet draught, 44 tons displacement, and was completed in 1895.

The application of turbines to marine propulsion became very rapid, and the construction of the turbine as applied to the larger ships called for the greatest skill of the designer, in regard to which no words of the author could pay adequate tribute. In 1904 turbine marine propulsion had reached such stages that by the end of that year some 26 vessels, aggregating 147,000 H.P. were in service, which created enough interest to lead George Westinghouse to consider entering the marine field.

At his instigation the late Admiral G. W. Melville, U. S. N., and Mr. John H. Macalpine, visited Europe in 1904 to report on the marine situation as it then existed, and it is interesting to note that, in spite of the large number of vessels equipped in Europe, the report was anything but encouraging. They did not see any opportunity for the steam turbine for driving ships as compared with the existing reciprocating engines, particularly in the case of ships which are required to operate at cruising speeds, and they showed how inferior were the turbine installations as compared with the reciprocating-engine installations when so operated. They recognized the applicability of the turbine to express steamers which always operated at their full speed, and pointed out their lack of economy when applied to lower-speed ships which comprise the greater part of the merchant marine. They said, "If one could devise a means of reconciling in a practical manner the necessary high speed of revolution of the turbine with the comparatively low speed of revolution required by an efficient propeller, the problem would be solved and the turbine would practically wipe out the reciprocating engine for the propulsion of ships. The solution of this problem would be a stroke of great genius," and proceeded some time thereafter themselves to become the genii by designing a tooth reduction gear adaptable to such service, embodying the now well-known floating frame. Mr. Westinghouse at once recognized the great possibility of its construction, and though considerably delayed on account of the financial depression at that time, a reduction gear expected to develop 6,000 H.P., reducing from 1,500 r.p.m. to 300 r.p.m. was tested in 1909.

The essential features of the floating frame gear will be but little referred to here in view of the many articles that have been written on this subject by Mr. McAlpine.†

---

*Report, slightly abbreviated, read before the Society of Naval Architects and Marine Engineers at Philadelphia on November 14, 1918.

†*Vide* "London Engineering" dated September 17, 1909, May 5, 12, 19, 26 and June 2, 9 and 16, 1916; Proceedings of Engineers' Society of Western Pennsylvania dated December 18, 1917; Institution of Naval Architects, London, March 29, 1917.

The point most freely discussed in connection with reduction gears is the permissible tooth pressure per lineal inch of tooth face. Unquestionably the safe load which can be transmitted by the gear teeth is all-important, but unfortunately the subject has been frequently regarded as entirely independent of any other feature of the reduction-gear design. This is a mistaken view. The ultimate limit of pressure which the teeth can safely carry has not been determined, but the pressures at present in use among designers are only possible with the most accurate alignment. No comparison of tooth pressure is therefore reliable without careful consideration of the methods used to maintain the unit pressure within the limits for which the design was made. Faulty bearing design, inadequate support or improper couplings may be individually or collectively responsible for deranging the alignment, resulting in a concentration of the tooth load and often failure of the teeth. It is obvious, then, that merely to limit the unit tooth pressure to some fixed empirical figure for all gears in no way insures the safety of the teeth. On the other hand, anything which tends to maintain the alignment of the teeth and to prevent concentration of pressure is to be welcomed as a source of safety in the operation of the gearing. It is for this reason that the Melville-Macalpine invention of the I-beam support for the pinion bearings is so thoroughly justifying the claims made for it. The closer this mechanically-correct device is investigated, the more apparent becomes its most important function, namely, to maintain automatically a very uniform distribution of tooth pressure under the most severe distortions of the gear-case that can occur in the frailest hull. Its value in permitting higher unit loads than are customary on which have been termed "rigid bearing" gears is graphically illustrated in Fig. 1. subjoined. The curves of the rigid-bearing dimensions have been com-

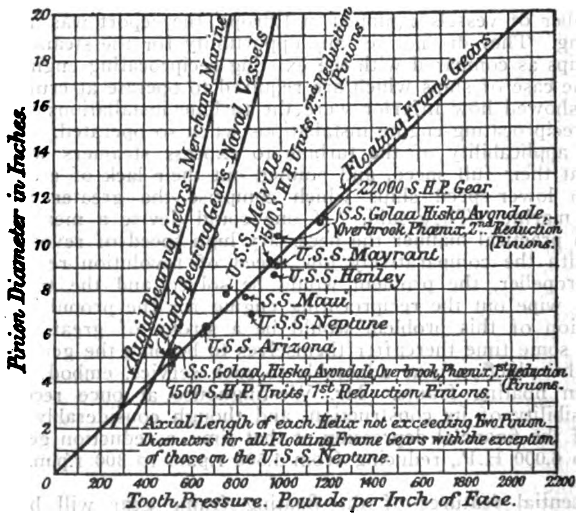


FIG. 1.

piled from recent marine practice in Great Britain. The floating-frame figures are those used by the Westinghouse Electric and Manufacturing Company for marine work, and there are points indicated on this curve showing gears actually in service.

As previously referred to, the gear by permitting high turbine speed, much simplifies the turbine design. It enables a speed to be selected for the turbine approprate for best economy. Furthermore, it inherently permits a design of increased reliability by virtue of sensibly reduced dimensions. The small high-speed geared turbines require few of the elaborate precautions in warming up and getting in service that must be practiced with the large direct-connected turbines.

Gears may readily be arranged with two pinions for even the smallest single-screw steamer, permitting the employment of what are known as cross-compound turbines; that is, a high-pressure and a low-pressure turbine, the steam passing them in series. Piping is so arranged that by the manipulation of valves the high-pressure turbine may exhaust direct to the condenser, or the high-pressure turbine be isolated and the low-pressure turbine receive high-pressure steam direct. Thus, generally speaking, no turbine or gear accident is likely to so incapacitate the machinery that the vessel cannot reach port at reduced speeds. There is one record of a 10½ knot steamer, which, on account of breakage of one pinion, crossed the Atlantic with the low-pressure turbine alone, obtaining speeds as high as 9 knots.

The general arrangement of such an installation is shown in Fig. 2,

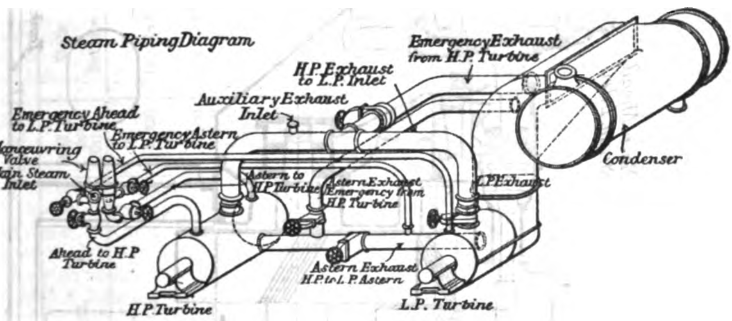


FIG. 2.

annexed. Its reliability has much to commend it. This is not only due to the flexibility above referred to, but further due to the simplification of the turbines themselves. Each turbine only expands the steam corresponding to half the heat drop, and so has a much reduced number of turbine elements, smaller temperature range, and reduced length between bearings as compared with a complete expansion turbine, having similar economy. Both high and low-pressure turbines, are provided with reversing elements arranged in a similarly compounded manner to the ahead elements, thus rendering maneuvering possibilities as complete with one turbine element operating as with both.

Of necessity the cross-compound turbine installation with its double-pinion gear is more costly than the single complete-expansion turbine, and with keen competition will not be installed unless shipowners recognize its manifest advantages. So far as the writer has observed, owners seem quite indifferent to the probable reliability of machinery so long as they secure insurance and the protection of the classification societies. The plea is therefore made by the author that underwriters, in collaboration with the classification societies, should discriminate in insurance rates between types of installations having manifestly differing degrees of reliability.

No doubt single-screw steamers will frequently be driven by single complete-expansion turbines, at least for low powers. An example of a small-sized installation, 1,500 H.P., is shown in Fig. 3, and attention is

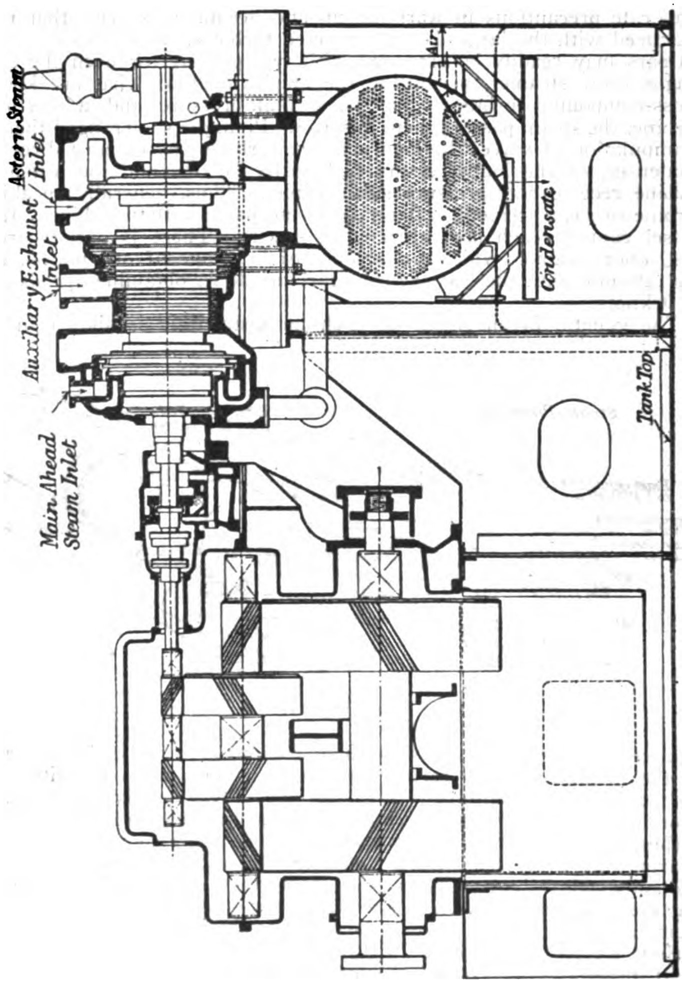


FIG. 3.

called to some of its features. The double reduction-gear elements are arranged one above the other with the high-speed pinion uppermost. This elevates the turbine axis and permits the condenser to be located immediately below the turbine, permitting the exhaust to issue from the underside of the turbine and pass vertically downwards to the condenser, much simplifying the removal of water. One end of the turbine is carried directly upon a bracket integral with the gear case, the other by the

TABLE II.

Name of Ship.	Where Built.	S.H.P. per Ship.	Date in Service.	Miles Travelled (approx.).	Troubles in Service.	Remarks.
Melville	New York Shipbuilding Co.	4,000	11- 1-15	7,000	None	U.S. T.B.D., tender main gear only.
Neptune	Maryland Steel Co.	6,500	12- 1-15	128,000	See Note A	U.S. collier.
Malmanger	Chester Shipbuilding Co.	2,900	3- 8-17	2,500	None	Sunk 3-23-17.
Westwood	Ames Shipbuilding Co.	2,500	3-24-18	16,000	None	
Maui	Union Iron Works	10,000	4-17-17	60,000	See Note B	
Golaa	Chester Shipbuilding Co.	2,900	7- 6-17	36,000	See Note C	
Hisko	Chester Shipbuilding Co.	2,900	12- 4-17	25,000	None	
Coronado	Moore Shipbuilding Co.	2,400	1-29-18	27,000	None	
Yosemite	Moore Shipbuilding Co.	2,400	2- 2-18	27,000	None	
Westerley	Ames Shipbuilding and Dry Dock Co.	2,500	2-27-18	7,600	None	Sunk 4-29-18.
Sudbury	Chester Shipbuilding Co.	2,900	3- 5-18	15,000	See Note D	
West Eagle	Ames Shipbuilding and Dry Dock Co.	2,500	4-12-18	11,000	None	
Yellowstone	Moore Shipbuilding Co.	2,400	4-25-18	22,000	None	
Overbrook	Chester Shipbuilding Co.	2,900	4-30-18	16,000	None	
Oakland	Moore Shipbuilding Co.	2,750	5-30-18	12,000	None	
Accomac	Los Angeles Shipbuilding Co.	3,000	6- 6-18	12,000	None	
Avondale	Chester Shipbuilding Co.	2,900	6-23-18	6,000	See Note E	
Wakulla	Los Angeles Shipbuilding Co.	3,000	6-26-18	—	None	
West Ford	Ames Shipbuilding and Dry Dock Co.	2,750	6-30-18	8,000	See Note F	
Wampum	Los Angeles Shipbuilding Co.	3,000	7- 9-18	—	None	
Wassiac	Los Angeles Shipbuilding Co.	3,000	7-25-18	—	None	
Montrolite	Ames Shipbuilding and Dry Dock Co.	2,500	8-12-18	5,000	None	
West Galoc	Los Angeles Shipbuilding Co.	3,000	8-13-18	—	None	
West Gleata	Los Angeles Shipbuilding Co.	3,000	9- 5-18	—	None	
Polar Sea	Baltimore Dry Dock and Shipbuilding Co.	1,800	9- 6-18	—	None	
Phœnix	Chester Shipbuilding Co.	2,900	9- 7-18	—	None	
Pennsylvania	Newport News Shipbuilding Co.	3,200	Not known	Not known	None	U.S. battleship cruising gears only.
Arizona	New York Navy Yard	4,000	Not known	Not known	See Note G	„ „
Mississippi	Newport News Shipbuilding Co.	3,000	Not known	Not known	None	„ „
Henley	Re-engined at Philadelphia Navy Yard	14,000	Not known	Not known	None	U.S. T.B.D.
Mayrant	Re-engined at Philadelphia Navy Yard	13,500	Not known	Not known	None	U.S. T.B.D.

exhaust outlet being set directly upon the condenser. The condenser is secured by its upper surface to seatings, eliminating disturbances to alignment on account of expansion of the condenser.

There appears to be much apprehension in the matter of the reliability of turbine-geared steamers, and if current rumor may be relied upon there have been many failures, which, with more careful design or by employing the best in the art, would seem to have been wholly unnecessary. To correct such an impression, Table II gives a list of all vessels equipped with the floating-frame gear and Westinghouse turbines. The table states the date on which the vessel entered service, the power of the installation, and approximately the miles traveled up to the present time (September 1, 1918).

Some troubles have been experienced as are certain to occur with all kinds of machinery. Since there must be a large human element, footnotes are appended, plainly stating the character of the troubles.

Note A.—U. S. S. *Neptune.*—This ship was equipped with two sets (twin screws) single-reduction gears and turbines, the turbines having been largely experimental and were not successful. The gears, however, operated splendidly. The turbine machinery was then replaced, and inasmuch as a higher turbine speed was considered desirable, new gears involving a greater speed reduction were also installed at this time. The designed constants of the new gears were very high and are still much in advance of our present practice.

Considerable trouble was experienced due to the teeth abrading which was only overcome by the use of lard oil. Since that time, however, in November, 1915, no further trouble has been recorded, and the teeth at the last inspection were in good condition without any indication of wear.

Note B.—S. S. *Maui.*—This vessel was built at the Union Iron Works. Inasmuch as we were late in the delivering the machinery, we were unable to carry out any tests on the gears in the works, and inasmuch as great store was set by the owners that nothing should go wrong on the maiden voyage, and as there was some uncertainty as to the excellence of the tooth surfaces, lard oil was employed as a lubricant for the first two voyages, after which mineral oil was substituted for the lard oil.

The operation of the machinery was all that could be desired for eight voyages, at which time there was abrasion of the gears due to failure of the lubrication. This was discovered in Honolulu and the ship returned to San Francisco with nothing being done to the machinery except an attempt to clean the oiling system. The tooth surfaces smoothed up considerably during the voyage.

The gears were then transposed from one side to the other so as to make what had been the astern surfaces the ahead surfaces, so as to use uninjured tooth surfaces for ahead operation.

Again because of some uncertainty of the tooth surfaces lard oil was employed as a lubricant and the vessel entered in Government service, going to Chile for cargo and proceeding to Baltimore. During the voyage there was an accident to an oil cooler, admitting salt water to the system. The shipbuilder having provided no means of removing any water, the effect of the salt water on the lard oil was to turn it violently acid, which necessitated a complete overhauling of the machinery on its arrival in Baltimore.

On this being completed and on the trip from Baltimore to New York trouble was experienced with abrasion of the tooth surfaces due, it was believed, to an inferior quality of oil, together with further failure of the oil cooler, admitting enough water to the system to make an emulsion of approximately 30 per cent. A better oil was substituted and nothing was done to the tooth surfaces, the vessel having made one voyage to Europe and back with no trouble as far as the gears are concerned.

During this voyage trouble was experienced with one of the turbines, causing quite an extensive repair because of an obstruction in one of the oil passages to the thrust bearing of the turbine, causing the thrust bearing to fail; the turbine rotor moving endwise, completely wrecking the labyrinth packing between the ahead and astern elements and buckling the rotor.

Note C.—S. S. *Golaa.*—Norwegian oil tanker built by Chester Shipbuilding Company, and chartered by British Admiralty. Put in service July 6, 1917. Made three round trips to English ports. At present in coastwise trade between Port Arthur, Texas and Philadelphia or Bayonne. Towards end of last trip from Europe (February, 1918) some teeth on starboard high-speed pinion broke out. This gear was disconnected and ship made port, using low-pressure turbine only. Inspection showed break occurred on one helix, 3 inches from end of tooth face. There was no evidence of wear on any of the tooth faces, the undamaged portion of broken tooth face indicating uniform distribution of load. This fact and the position of fracture make certain accident was due to defective material.

In August of this year the high-pressure turbine suffered injury as a result of improper setting of the turbine thrust bearing. Shortly thereafter an accident occurred to the pinion which replaced the one originally defective. The character of the broken tooth was similar to the previous one. The broken parts have not yet been received to enable a careful examination of the material to be made.

Note D.—S. S. *Sudbury.*—An American freighter built by the Chester Shipbuilding Company for the Shawmut Steamship Company. It is at present being operated by the United States Naval Overseas Transportation Service. A few days after leaving France on return half of maiden voyage, some teeth broke out of starboard high-speed pinion. This gear was disconnected and ship returned to New York, using low-pressure turbine only, with which 9 knots was attained, the designed full speed being 10½ knots.

On arrival at port, inspection showed no wear, the tooth load having been well distributed on all tooth faces. Several teeth were cracked. Subsequent examination of steel indicates defective material, most probably due to faulty heat treatment, the fracture indicating the steel had been burned. This was confirmed by discovery that some teeth in port high-speed pinion were also cracked, although no failure occurred in service. Both pinions were forged from the same billet and treated at the same time. These pinions are similar as to material and design with 25 other pinions now in successful operation.

Note E.—S. S. *Avondale.*—An American oil tanker, built by the Chester Shipbuilding Company, and being operated by the Pan-American Oil Company for the United States Shipping Board. Just as ship was leaving Philadelphia for trial trip a heavy blade rub was heard in high-pressure turbine. Inspection indicated defective workmanship. Some of the rotating blades were loose as a result of having been improperly put in. The spindle was rebladed and ship proceeded satisfactorily from Philadelphia to New York. This turbine is a counterpart of 12 others now in successful operation on various ships.

Note E.—S. S. *West Ford.*—This vessel left Seattle at 5 P. M. on July 14, 1918, on her maiden voyage to an Atlantic port. Seven hours out from Seattle, oil commenced foaming out from the turbine, running over the engine-room gratings. The chief engineer was called and, instead of investigating the reason for augmentation of the oil, he was content to pump 200 gallons to a reserve tank. Later the same thing occurred, oil again running out on the gratings, and later still at 2.14 A. M., the high-pressure turbine commenced vibrating so badly that it interfered with their lighting system. They shut down to remedy the matter of the lighting

system, and on trying to start the turbine it required 125 pounds initial pressure to revolve it. Serious vibration was again in evidence, so the chief decided to do what he termed "crank up on the thrust." It was impracticable to anchor, so they returned to Port Townsend at 40 r.p.m. They then made an observation of the oil in the drain tank, and on removing the cap the oil squirted to the roof of the shaft alley. It was later discovered an oil cooler tube had split, admitting large quantities of salt water to the oil system.

Instructions had been given that the level of the oil in the drain tank should be regularly observed, and it is incomprehensible that it was not at once obvious that something was getting in the oil system from some exterior source. Merely looking at the oil would have determined this was salt water. Then all that would have been necessary was to shut off the water service to the cooler and separate the relatively small quantity of salt water from the gravity tank, and there need have been no interruption whatsoever. The result of such neglect was for the bearings, for want of lubrication, to let the rotor down, causing considerable injury to the turbine.

Note G.—U. S. S. *Arizona.*—On a voyage from Cuba to the United States the teeth of one of the cruising gears scored. Subsequent investigation showed that the oil pumps had stopped and the supply to the gears had failed. The teeth were dressed up and no further trouble has been experienced.

As in land installations, the overall economic performance is not dependent on the prime mover alone. High performance is not to be secured without care being bestowed on all the auxiliary apparatus, attention to matters of heat balance with the feed-water heater, condensing apparatus capable of performing with a close approach to the ideal, &c.

Large size land turbines have been built which deliver to the shaft not less than about 80 per cent of the theoretical energy available from the steam expanding between the limits specified, so that further improvements in the turbine itself will not materially raise this efficiency. Further improvement in economy must be looked for from causes other than the prime mover itself. This is a subject of the greatest importance in view of the rapidly increasing cost of fuel, and it justifies considerably more capital expenditure for economizers and other such apparatus which will reduce fuel cost.

The engineer of our large central stations is subject to keen competition and is confronted almost solely with the problem of producing reliable energy for the least cost, including fixed charges. He has before him daily statements showing his overall power costs with a high degree of accuracy, resulting in considerable success on his part in the production of low power costs.

It is therefore not unreasonable that the marine engineer might look to the central-station engineer for advancement from some of the latter's best practice. It is, of course, understood that complete standby machinery is not available for the marine engineer, neither can the superintending engineer direct those in immediate charge of the machinery by frequent visits. Hence there is the more urgent call for the greatest reliability and simplicity of the marine installation; though, after all, perhaps not more than that which the central stations have in our big cities. An interruption to service, however brief, is regarded most seriously.

Feed-Water Heating Systems.—It is common practice on shipboard to provide a closed or tubular heater to which is led the exhaust steam from the auxiliary machinery, which latter exhausts at a pressure between 5 pounds and 10 pounds gage. The exhaust system is provided with a connection to let any proportion of this steam to the condenser. Sometimes again there is a hand valve, permitting any surplus of this steam to be

taken to a low-pressure stage of the turbine, all of which calls for hand manipulation where the conditions are sometimes quite variable, owing to more or less intermittent operation of the feed pump. Sometimes the connection to the condenser is provided with a spring-loaded valve so that when the exhaust pressure reaches a predetermined pressure, higher than that within the condenser, steam may pass to the condenser. The construction of the valves usually employed is such that the higher the pressure in the condenser, the higher is the pressure at which the steam will pass to the condenser, which is, of course, undesirable. A system which will automatically maintain a predetermined temperature in the feed heater at all times, bleeding the main turbine for this purpose, at high powers if necessary, and at the same time automatically permit any surplus steam not condensed in the feed-water heater to do useful work in the turbine, is much to be desired.

Heat balance systems have for some time been available for land purposes, there being provided automatic valves so that whenever there is a surplus of auxiliary exhaust steam this surplus may pass through a low-pressure stage of the turbine, doing work therein at the rate of 20 pounds to 22 pounds per horsepower. Other types of valves are available for land installations which will not only take surplus steam into the turbine at times of light loads on the main unit at which time the auxiliaries will obviously furnish too much exhaust steam; but, on the other hand, they are arranged to deliver steam from the turbine to the feed-water heater at times of heavy loads when the stage pressures in the turbine are relatively high and there is a deficiency of steam for feed-water heating. The latter should particularly find a place on war vessels which are

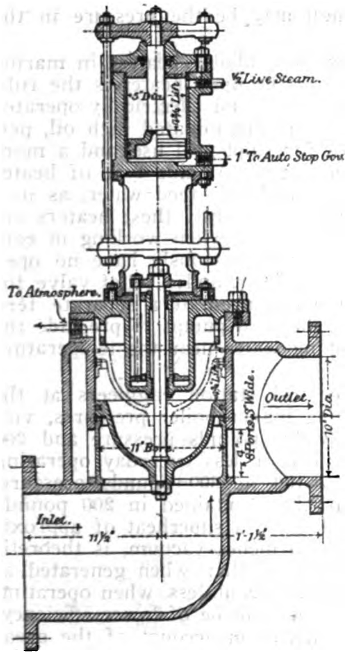


FIG. 4.

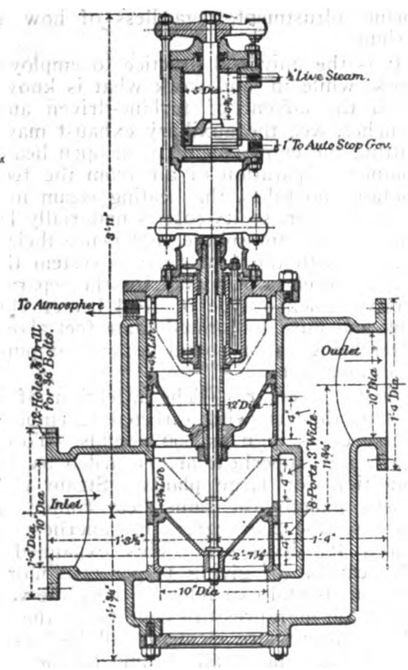


FIG. 5.

required to operate for long periods at different speeds, where there is a widely varying ratio of steam used by the auxiliary machinery to that used by the main turbines. In merchant vessels this ratio is more nearly constant, but nevertheless careful study to the end of obtaining a complete heat balance will be advantageous. Details of the two forms of valves above described are shown in Fig. 4 and Fig. 5, subjoined.

The valve shown in Fig. 4 is intended for cases where there is never a deficiency of auxiliary exhaust steam for feed heating. It comprises a piston having atmospheric pressure above it, and the auxiliary exhaust steam below. When the auxiliary exhaust steam pressure rises a sufficient amount above the pressure of the atmosphere to raise the piston, steam may pass to the turbine. Air leakage into the turbine is entirely precluded by ample steam seal around the piston furnished by the auxiliary exhaust steam. The piston may be loaded to any feed-heater pressure desired. The upper cylinder is to forcibly close the valve in the event of the turbine over speeding.

The modification of this valve shown in Fig. 5 is adaptable where there are at times a deficiency of auxiliary exhaust steam. Any time the stage pressure in the turbine is greater than that in the heater the lower piston will raise, permitting passage of steam from the turbine to the heater at the same time, when required, permitting flow to the turbine as in the case of the valve shown in Fig. 9.

It sometimes occurs that the turbine is subjected to such overloads that a stage of the turbine selected to give the desired heater pressure at moderate loads would be far too high for the heater at overloads. In this instance the modification shown in Fig. 6, subjoined, may be employed which limits the pressure in the heating system in accordance with the spring adjustment, regardless of how high may be the pressure in the turbine.

It is the universal practice to employ closed tubular heaters in marine work, while in land work what is known as the open heater is the rule. With the advent of turbine-driven auxiliaries and electrically-operated winches, &c., the auxiliary exhaust may be uncontaminated with oil, permitting the employment of an open heater for shipboard use, and a more complete separation of air from the feed water. As this type of heater contains no tubes, the heating steam mixes with the feed water, as in a jet condenser, so its cost is materially less. Frequently these heaters are open to the atmosphere, and hence their name, but when working in connection with the heat balance system they must obviously have no open outlet, merely a provision for the separation of air, and a relief valve for safety. Such a type of feed heater eliminates need of a separate feed tank and must be located some feet above the feed pumps to preclude the latter being vapor bound because of pumping water near the temperature of evaporation.

Steam Pressures.—The attention of central station engineers at the present time is being directed to employing higher boiler pressures, viz., pressures as high as 600 pounds. To-day 200 pounds pressure and 200 degrees F. superheat are regarded as a more or less everyday operating condition for large plants. Steam generated at 600 pounds pressure, having exactly the same heat content as that contained in 200 pounds pressure and 200 degrees F. superheat will have a superheat of approximately 128 degrees F. This, expanded to 29 inches vacuum, is theoretically capable of giving 13 per cent more energy than when generated at 200 pounds and expanded to the same vacuum. Doubtless, when operating under the high-pressure conditions, the turbine will be of lower efficiency. The high-pressure element will be less efficient on account of the great density and the small volume of the steam, and on the other hand the low-pressure elements will be less efficient because of the great amount of

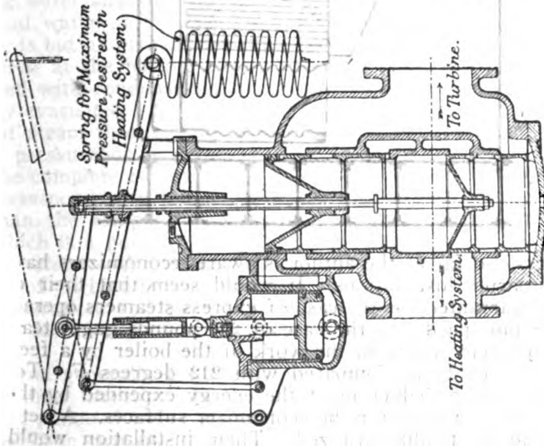


FIG. 6.

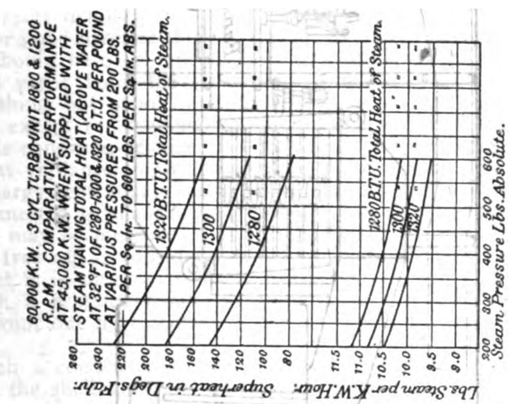


FIG. 7.

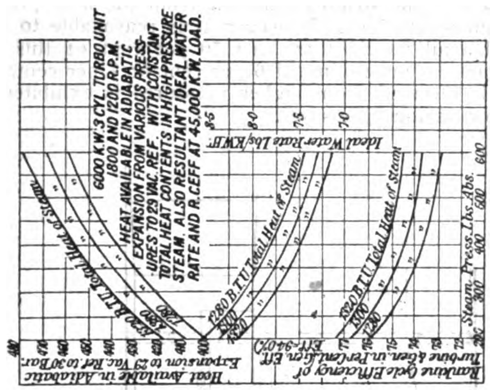


FIG. 8.

water precipitated by the steam expansion from the high pressure, introducing a brake in the turbine. However, it is reasonable to suppose that the turbine will avail itself of at least 50 per cent of this 13 per cent possibilty, producing a net saving of 6 per cent or 7 per cent. What may be expected to be derived from higher pressures is exhibited in Figs. 7 and 8, plotted for various pressures.

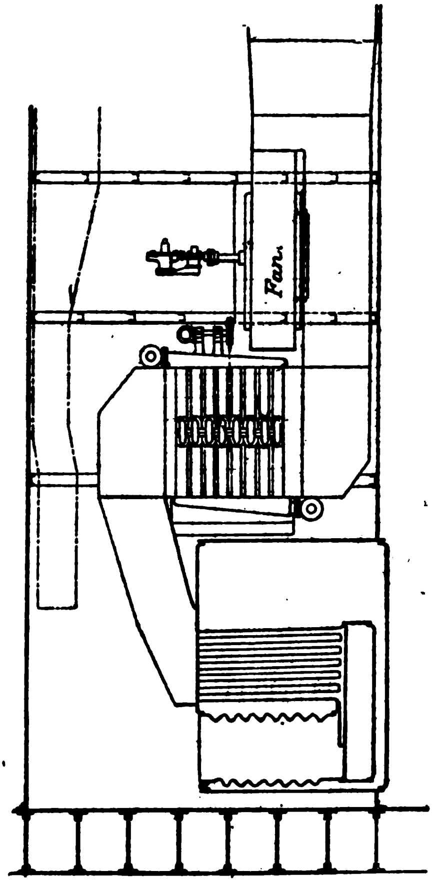


FIG. 9.

Economizers.—So far as the author is aware, economizers have not been employed in marine installations. It would seem that their capital cost would be well warranted for the case of express steamers operating always at their maximum speed. In the case of 250 pounds gage steam pressure there is a 16 per cent saving in the work of the boiler by a feed temperature of 375 degrees F. as compared with 212 degrees F. To this gain there must be made a deduction of the energy expended by the induced-draft fans and the scrapers for the economizer surfaces. A net gain of 10 per cent should be readily realized. Their installation would seem to present no particular difficulty as is shown in Fig. 9.

Superheat.—With the old direct-connected turbines, superheat has never been recommended, and wisely so considering their immense structure. Even with the much smaller high-speed geared turbines as high superheat as is desirable in land installations may not be practicable at sea on account of the rapid stopping, starting, &c., necessary for quick maneuvering, which may cause uneven heating and cooling. In the author's opinion superheats as high as 100 degrees F. should present no operating difficulty whatsoever with well designed high-speed turbines. One hundred degrees F. superheat will affect the steam consumption not less than 8 per cent as compared with dry saturated steam, probably saving some 4 per cent or 5 per cent in fuel.

Condensers.—Inasmuch as turbines may be designed to expand the steam to vacuum of 29 inches, and as the steam consumption with such a turbine will be 7 per cent better with 29 inches vacuum than with 28 inches providing it be designed to expand to such low pressures, the necessity of providing sufficient well-disposed surface to produce such a vacuum is not only obvious, but it is furthermore necessary to have a complete understanding of what constitutes good condenser performance with the apparatus available.

An important detail is the drop in steam pressure through the condenser. For good practice this should not exceed 1/10 inch mercury. The air pump should be of a type capable of completely evacuating the non-condensible vapor from the condenser so that the condenser shell shall contain nothing but steam, and therefore be of the same temperature throughout except for the small pressure drop within the shell itself. To produce this condition requires not only excellent air-removing apparatus, but constant vigilance in the elimination of air leaks. It is the practice of certain large central stations to provide an air bell, which may on occasion be connected to the discharge of air pumps, by means of which the air leakage may be measured. With 40,000 H.P. units, 3½ cubic feet per minute of free air is considered good practice, and should this reach to 7 cubic feet it is the signal for a search to discover the source of the leak. Such a practice would be impracticable on shipboard and only serves to point out how central-station engineers regard the ill-effects of air leaks.

With such a condenser having practically uniform steam temperature throughout the shell, there is the greatest temperature difference between circulating water and steam in the bottom pass (assuming steam enters at the top and water at the bottom); while on the other hand, in the top pass there is but a small temperature difference. Hence much more steam is condensed at the bottom than at the top, contrary to what is usually experienced with condensers that do not have the non-condensible vapors completely evacuated. This calls for especial care in having paths for the passage of steam through the upper zones of the condenser to insure the minimum pressure drop.

With the complete evacuation of the non-condensible vapors, the vacuum should correspond to a temperature not to exceed 1 degree or 2 degrees higher than the temperature of the condensed steam leaving the condenser, which is a point that should be kept constantly under observation.

Oiling Systems.—While the oiling system can bear no relation to the economics of turbine-propelled ships, it is of vital concern to the reliability of operation, and too much care cannot be given to the lay-out of the system to the end of obtaining the extreme degree of simplicity and reliability.

It has been generally the custom in Westinghouse installations to provide oil pumps directly driven by gearing connected to the main gears. The double-pinion double-reduction gears, of which a large number have been bulit, have two oil pumps, one driven from each intermediate gear

shaft. The design contemplated for the sake of reliability that each oil pump alone be large enough for the system. These pumps are relayed by an auxiliary steam-driven pump for use when maneuvering. The direct-driven pumps have been objected to in some quarters and duplicate steam-driven pumps substituted therefor. The advantage of the latter, while no doubt lacking the extreme reliability of the direct-driven pumps, is that, when properly controlled, they only pump the oil needed by the system instead of, say, three times as much, which no doubt will have much to do with increasing the life of the oil as well as simplifying the filtering or straining systems. What is regarded as an ideal system is shown in Fig. 10. It has the following features:

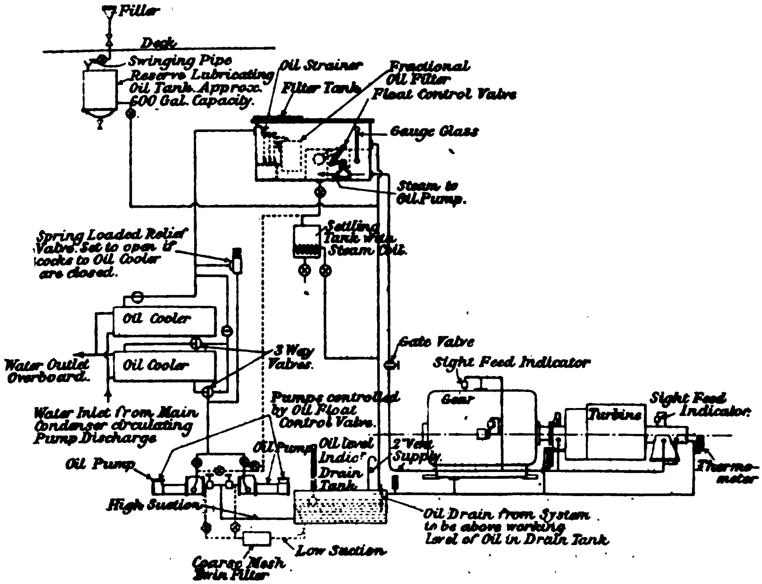


FIG. 10.

1. The pumps are located at as low a level as possible to reduce the suction head to the extreme limit.

2. The drain tank is of as large a dimension as practicable, and is located sufficiently low to insure the oil draining freely to it from the machinery with ample-sized pipes, and assurance that with the extreme rolling and pitching there will be no leakage from the bearings.

3. The pump suction should be several inches above the bottom of the tank to avoid drawing dirt and emulsion from the bottom.

4. It is regarded as desirable to employ an additional suction from the bottom of the tank by means of which dirty oil may be pumped through a filter and to permit complete emptying of the tank for cleaning purposes.

5. The pump is arranged to discharge its oil through coolers to an overhead gravity tank located some 20 feet above the machinery.

6. Duplicate coolers are employed so that while one set is in use the other may be cut out of service. They should be by-passed with a spring-loaded check valve, loaded to some amount greater than the resistance of the cooler, so that under no conditions can mishandling of the valves cause interruption to service.

7. An overhead gravity tank is provided with three strainers having successively finer mesh, the first being sheet metal with 1/16 inch perforations, the second 30-mesh wire gauze, the last 70-mesh wire gauze.

In the event of these becoming choked there can be no interruption to service, for the oil mostly flows over the top, when they may be readily removed for cleaning. The screens are secured to quite large cast frames arranged so that any dirt falling from the screen on their being removed will be caught in the frames and thus removed.

From these the oil passes through a fractional filter, that is, some of the oil runs through filter bags and is thoroughly filtered, the rest overflowing. The gravity tank should be provided with a very rugged float of large power for controlling the speed of the oil pumps. The outlet from the tank to the machinery should be well above the bottom and there should be other means of drawing oil from the bottom of this tank to a filter or settling tank. The character of the oil in the bottom of this gravity tank should be regularly observed to be sure that the water is not entering the system from any source.

8. The system which provides a settling tank capable of taking a complete charge of oil for the system is to be recommended.

9. It is considered desirable that the oil coolers should be arranged, if possible, to have the oil pressure superior to the water pressure, so that a loss of oil will occur rather than the admixture of salt water. The former would be shown by the oil level indicator which is situated in the lower tank.

There are installations, such as destroyers, where there is but little head room and where a gravity-oiling system is impracticable, and a pressure-oiling system must be employed. In this case twin strainers or their equivalent must be resorted to. These are, of course, dangerous because of them becoming choked and interrupting the service, or bursting and letting dirt through if the pumps are sufficiently powerful. If they must be employed they should invariably be by-passed through a spring-loaded check valve, loaded to, say, ½ pound, so that with a slight increase of resistance of the strainers, the oil will by-pass them and the service never be interrupted.

Concerning arrangements of the turbines themselves, a design of a single complete expansion unit has already been referred to. The general design of the cross-compound units are shown in Fig. 11, and Fig. 12. The receiver pressure between the high and low-pressure elements approximates 5 pounds gage at full power, rendering it adaptable for connection to the auxiliary exhaust steam and the feed heater as has previously been discussed. The detail construction of these turbines, method of inserting blades, &c., &c., will not be dealt with in detail here as already much has been written concerning them,* except those details which apply particularly to marine work.

The turbines illustrated in Figs. 11 and 12 are essentially for vessels always operating at their maximum speed and are subject to modifications on ships where cruising speeds are involved. One of these, known as the "divided-flow" type, is shown in Figs. 13 and 14 and are the turbines applied to a Swedish cruiser. The combination comprises an impulse element in the high-pressure turbine through which passes all the steam to the system. The nozzles are arranged in groups, each under valve control so that the turbine may realize substantially full pressure at the nozzles with widely differing volumes of steam flow. The general arrangement of this combination is shown in Fig. 15. After passing the impulse element the steam divides, one portion continuing through the same turbine to condenser pressure; and a slightly greater portion passing over to the low-pressure element through which it expands to the condenser. For cruising

*See "Electric Journal" for January, February, March, April, 1918.

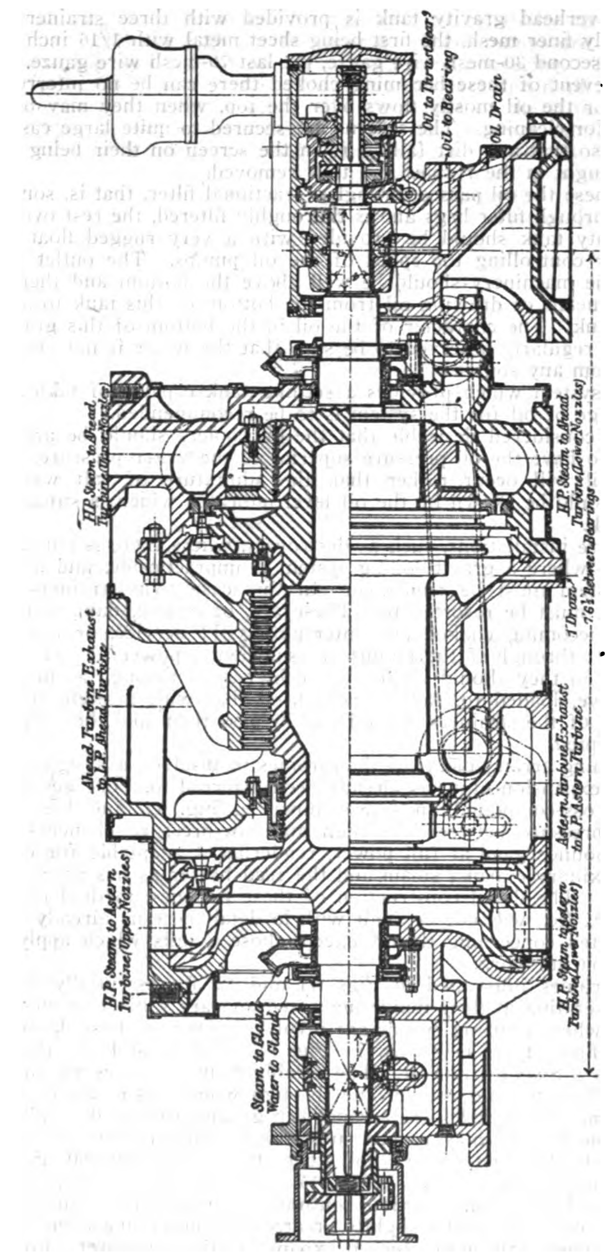

Oil to Thrust Bear'g
Oil to Bearing
Oil Drain
H.P. Steam to Ahead Turbine (Upper Nozzles)
Ahead Turbine Exhaust to L.P. Ahead Turbine
H.P. Steam to Astern Turbine (Upper Nozzles)
Steam to Gland
Water to Gland
H.P. Steam to Ahead Turbine (Lower Nozzles)
Drain
7' 6½" Between Bearings
Astern Turbine Exhaust to L.P. Astern Turbine
H.P. Steam to Astern Turbine (Lower Nozzles)

FIG. 11.

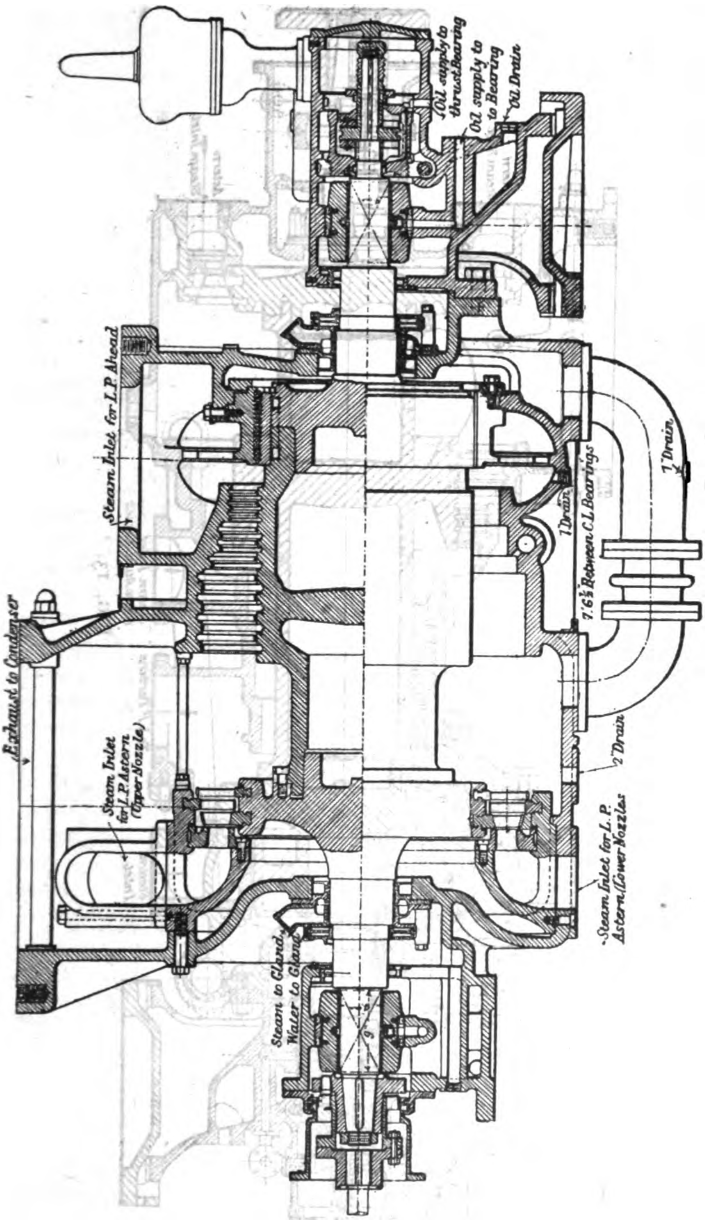


Fig. 72. L. P. ELEMENT OF CROSS-COMPOUND TURBINE

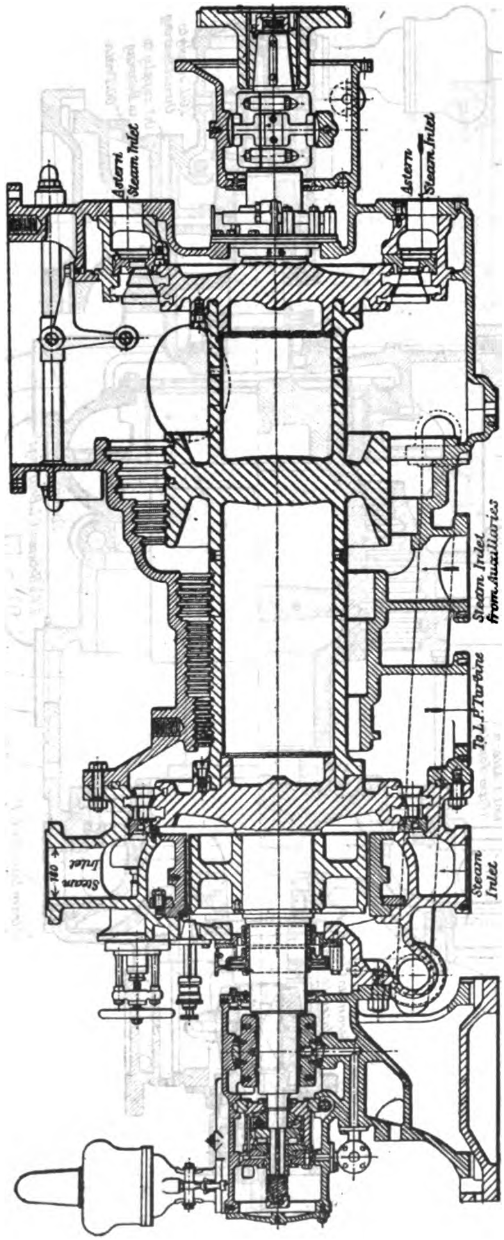
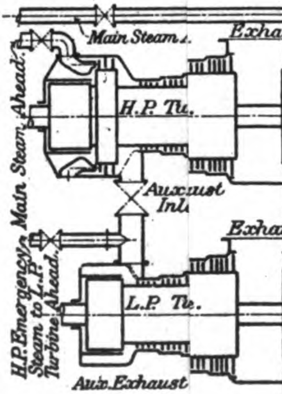

Astern
Steam Inlet
Astern
Steam Inlet
Steam Inlet
from Auxilliaries
To L.P. Turbine
Steam
Inlet
180
Steam
Inlet

FIG. 13.


Main Steam
Exha
H.P. Tu.
Auxaust
Inl
Exha
L.P. Tu.
Aux. Exhaust
Main Steam Ahead.
H.P. Emergency Main Steam to L.P. Turbine Ahead.

15.

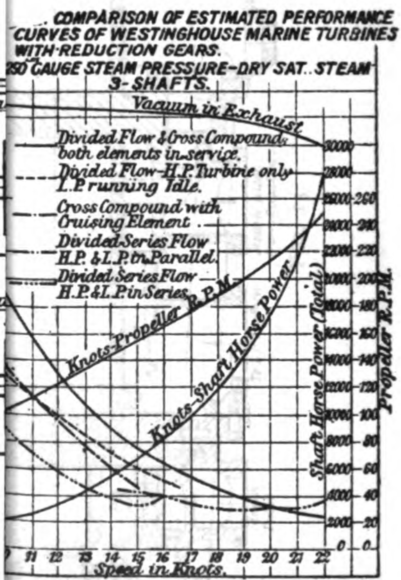

COMPARISON OF ESTIMATED PERFORMANCE
CURVES OF WESTINGHOUSE MARINE TURBINES
WITH REDUCTION GEARS.
250 GAUGE STEAM PRESSURE-DRY SAT. STEAM
3-SHAFTS.
Vacuum in Exhaust
Divided Flow & Cross Compound, both elements in service.
Divided Flow-H.P. Turbine only L.P. running Idle.
Cross Compound with Cruising Element.
Divided Series Flow H.P. & L.P. in Parallel.
Divided Series Flow H.P. & L.P. in Series.
Knots-Propeller R.P.M.
Knots-Shaft Horse Power
Shaft Horse Power (Total)
Propeller R.P.M.
Speed in Knots.

FIG. 17.

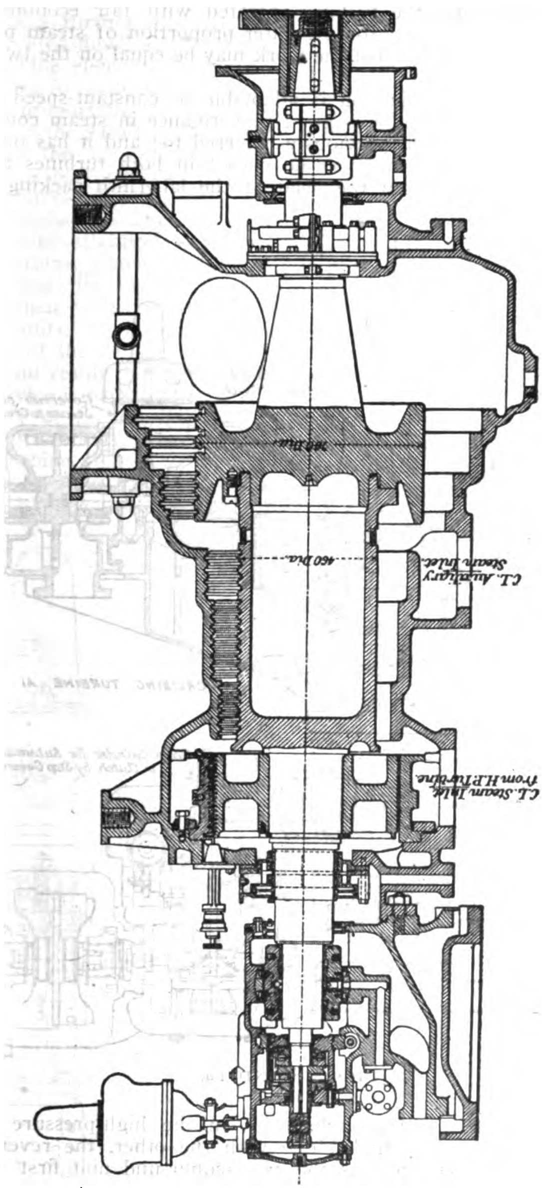
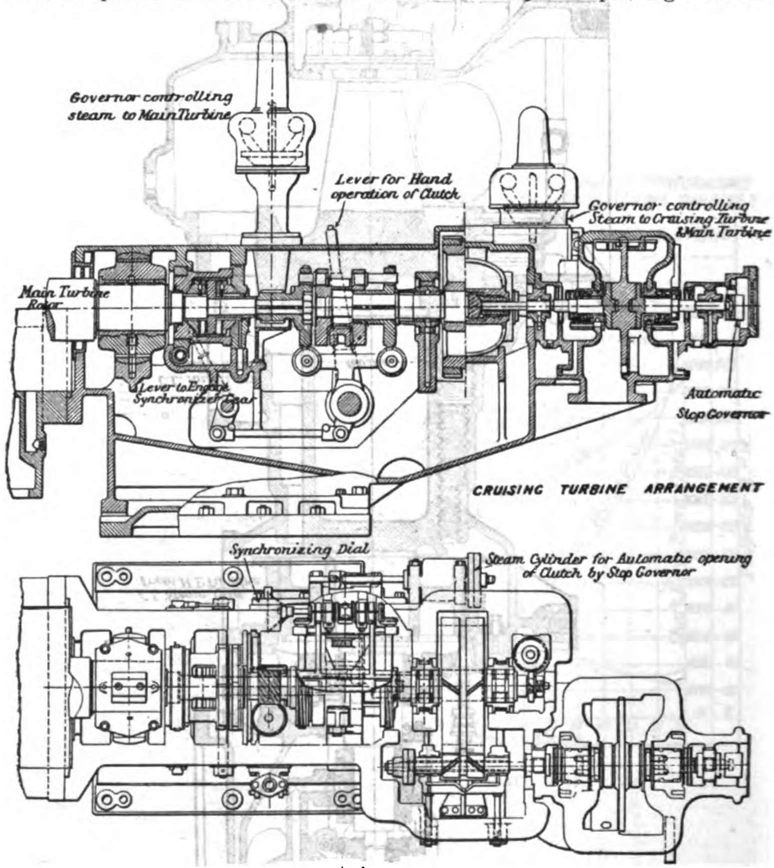

C.L. Auxiliary
Steam Inlet.
C.L. Steam Inlet
from H.P. Turbine.
460 Dia.

FIG. 14.

the one low-pressure element may be cut out of service, the steam closed to certain nozzles, and the turbine operated with fair economy at the reduced power. The reason for a greater proportion of steam passing to the low pressure is in order that the work may be equal on the two pinions at full power.

This divided flow combination is adaptable to constant-speed steamers and will generally give nearly as good performance in steam consumption as the cross-compound unit previously referred to; and it has one advantage, that reversing elements may be placed in both turbines as in the cross-compound unit without the need of the labyrinth packing between



FIGS. 18 AND 19.

the ahead and astern exhaust chambers in the high-pressure elements. Although not being compounded one with the other, the reversing elements are not as economical as the cross-compound unit first described. At full power the steam passes in parallel through three sets of low-pressure blades (if the low-pressure turbine be double flow), thus permitting smaller turbine diameters or higher rotative speeds, or both if desired, without reducing the adaptability of the turbine to expand to low

pressures. This type of installation has the advantage of smaller cross-connecting pipes than the straight cross-compound.

There is a further modification of these arrangements known as the "series divided flow" as illustrated in Fig. 16, which provides for certain of the turbine elements being operated in parallel for maximum powers and in series for reduced powers. It will be observed from the figure that the blade proportions work out with approximate corrections for either the series or the parallel condition of operation. This combination renders unnecessary the installation of a separate turbine for use when cruising. The relative performance of these respective types of turbines are exhibited in curves Fig. 17.

Some installations have been made where cruising turbines have been employed and attention is called to the arrangement for coupling the cruising turbine, shown in Fig. 18. It has been regarded in the past as desirable that the cruising turbines be arranged so that they may be operated when connected to the main turbines at the maximum speed of the main units, the idea having been that should the engineer go up to full power of the vessel and forget to disconnect this cruising turbine, no injury would result therefrom. This meant that when the cruising turbine was in service it had to operate at very low blade speeds and be therefore quite heavy or the cruising turbine cannot be arranged to be very economical. It is the practice of the Westinghouse Company to design the cruising turbine geared to the main turbine and designed for the speeds at which it is required in service which would be dangerously high should it be not disconnected at full power.

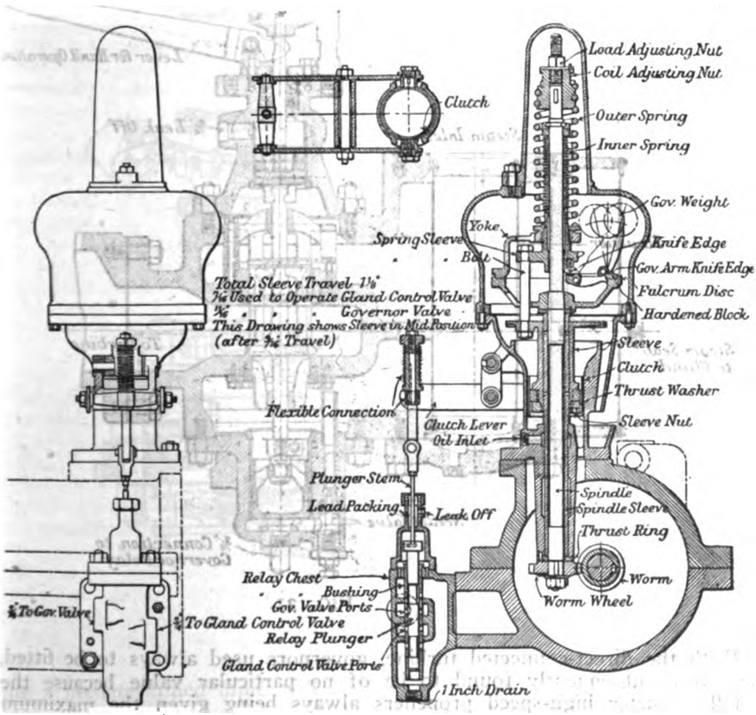


FIG. 20.

The design shown in Figs. 18 and 19 provides that should the cruising turbine operate above a predetermined speed, steam will automatically be shut off from it and the clutch automatically opened. The arrangement of this installation provides that the steam maneuvering valve is employed whether the cruising turbine is in operation or not. Means are provided for readily synchronizing the cruising turbine to the speed of the main turbine and throwing the clutch without affecting the speed of the vessel, provided it is running at speeds corresponding to that at which the cruising turbine should be operated.

Concerning steam turbine details that differ from land practice because of their application to ships, there are the following:

Governing Arrangements.—With the old reciprocating engines, racing in heavy weather was always to be reckoned with. While governing arrangements were sometimes furnished by the builder, they usually passed most of their existence in a store-room locker. While the reciprocator may accelerate rapidly, it may generally run at proportionately higher relative speeds than the turbine without injury.

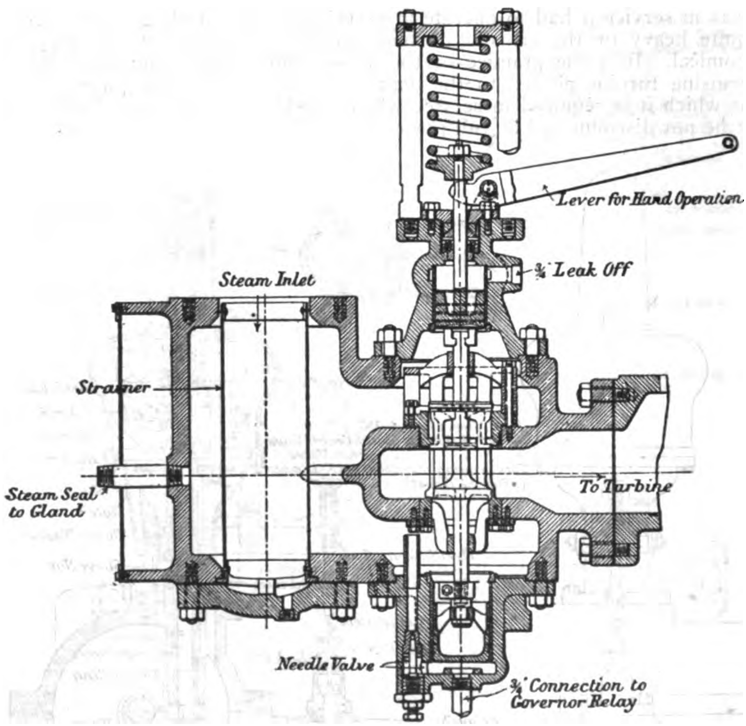


FIG. 21.

With the direct-connected turbine, governors used always to be fitted, but were subsequently found to be of no particular value because the small diameter high-speed propellers always being given the maximum submergence seldom came out of water with the vessel pitching. With the geared turbine the propeller dimensions revert to that of the recipro-

cating engine, and while, because of its inertia the turbine and gear may accelerate less rapidly than the reciprocator, very material increased speeds would be destructive. Therefore, a very dependable regulating mechanism is desirable which must at least have some semblance of being able to regulate, and must not be a mere stop governor which, on the speed reach-

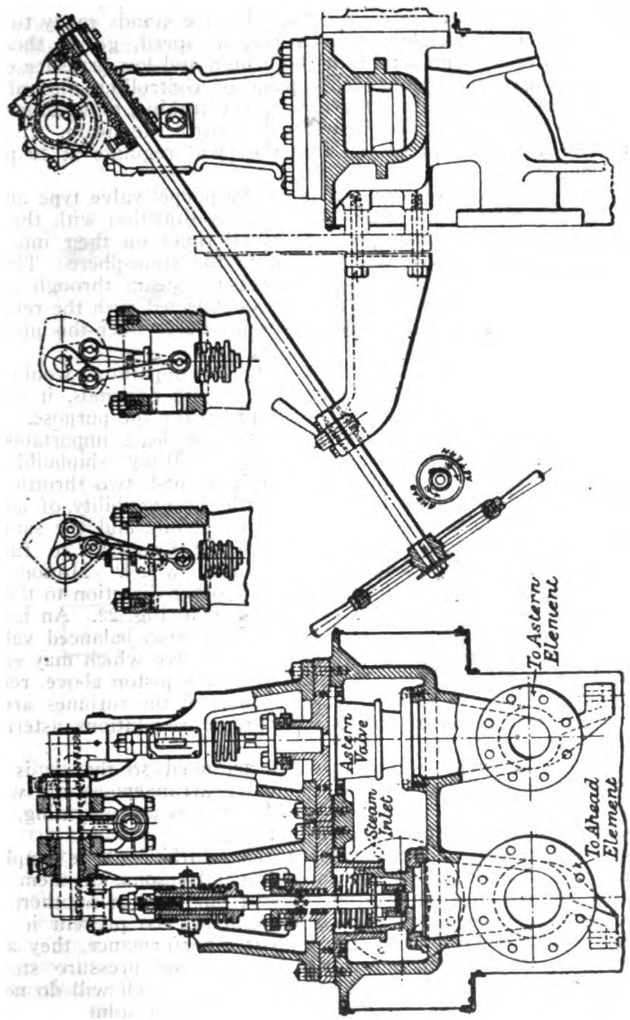


FIG. 22.

ing a predetermined limit, will slam shut an automatic valve. Arrangement of governing employed by the Westinghouse Company is exhibited in Figs. 20 and 21.

The governor is of the ordinary fly-ball type, the weights being carried on knife edges, and all working parts arranged to be under forced lubrication. The governor controls the valve shown in Fig. 21 by means of the steam relay, so that the connection between the turbine and the governor valves is merely a pipe, thus eliminating the necessity of connecting links, levers, etc. The relay may be moved by hand at any time by depressing the small spring that is above the relay. In this manner assurance may be obtained at any time without interfering with the operation of the vessel that the governor-control valve stands ready to control should the governor weights, with a rise in speed, go to their outer position. With cross-compound units both high and low-pressure elements are provided with a governor, each capable of controlling the valve, that is, either governor may reduce the flow to the turbine, or in other words, whichever governor is running fastest will control the steam. It is usual to adjust the governor springs so that they will regulate at 10 per cent above normal speed.

Concerning the governor valve, it is of the poppet-valve type and actuated by the two pistons above and below in conjunction with the spring. Both pistons have full steam pressure at all times on their inner sides. The outer side of the upper one is open to the atmosphere. The lower side of the lower piston receives high-pressure steam through a needle valve. The lower side of the piston is also connected with the relay adjacent to the governor which will control the pressure under this piston, and hence the position of the valve.

While the regulating characteristics of this apparatus would hardly meet the rigid requirements for turbines driving dynamos, it is simple and regulates with more than sufficient accuracy for the purpose.

Maneuvering Valves.—An important detail, at least important to the watch engineer, is the maneuvering valve. Many shipbuilders—for economy, we presume—are content to merely furnish two throttle valves, one for ahead and the other for astern, with the possibility of admitting steam to both the ahead and astern elements at one and the same time. It has been the practice of the Westinghouse Company to furnish a maneuvering valve operated by a single hand wheel. Motion in one direction gives steam to the ahead, and in the other direction to the astern elements. The design of this valve is as shown in Fig. 22. An important feature is that the valves themselves are single-disc, balanced valves, so that there is but a single ground seat for each valve which may easily be maintained bottle tight. They are balanced by the piston above, rendering their operation easy. With this valve mechanism the turbines are easily brought from full speed ahead to full turbine revolutions astern in 20 seconds, or *vice versa*.

In all cases the maneuvering valve is arranged so that it is always employed, no matter what may be the valve arrangements or what the method of operation; that is, whether both turbines are operating, or only one in case of disablement, as shown in Fig. 2.

Turbine Glands.—Most builders of marine turbines have employed a labyrinth type of packing for the turbine glands; some of them a combination of this with spring packing rings of one kind or another. There is no pretence of these glands being steamtight. To prevent ingress of air, which would interfere with the condenser performance, they are provided with a steam connection for admitting high-pressure steam for sealing so that there shall be a leakage of steam, which will do no harm, rather than one of air. It is found in practice that absolutely to preclude any air leakage there must be a superabundance of steam admitted which will blow, in considerable volumes, into the engine-room. A turbine gland used to be regarded as one of the difficult problems of design detail until, in 1903, the Westinghouse Company devised what is known as the water

gland. It merely comprises a centrifugal pump runner operating in a casing which, if furnished with water at the axis, would raise that water to some 10 pounds higher pressure than the maximum pressure against which the gland is designed to pack. In service its periphery is furnished with water at some 5 pounds greater pressure than the maximum pressure at which the gland is to pack, thus providing a water annulus at the outer part of the gland, forming a hermetic seal which is very effective. For condensing turbines a water pressure of 5 pounds gage at the periphery of the gland is satisfactory. This simple device rendered an exceedingly difficult detail one that could be forgotten, and obviously has its application to marine installations.

The design of this gland as applied to marine installations is shown in Fig. 23, in which there is a combination of the water gland above described

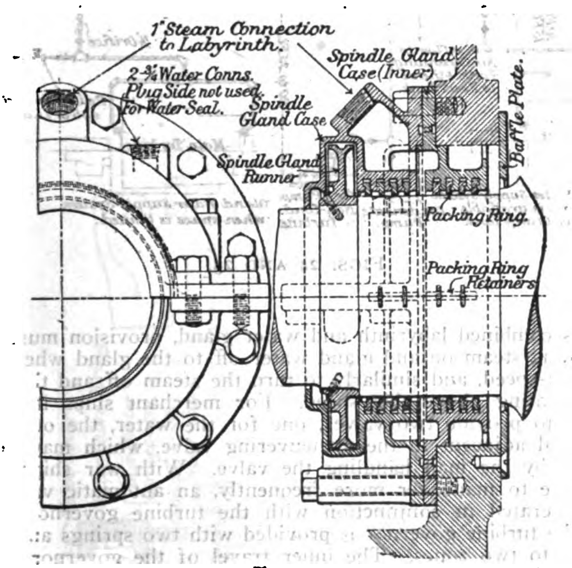


FIG. 23.

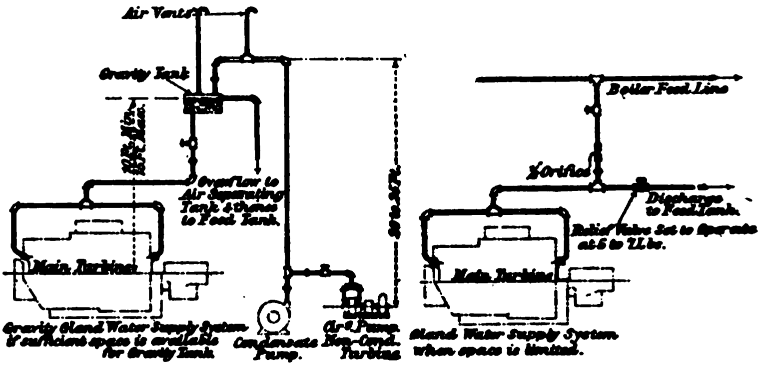
for use when operating at anything above half-speed, and combined therewith is a simple steam labyrinth for use when running at speeds below one-half or when standing by. This labyrinth is embryotic as compared with labyrinth glands as generally understood, and no attempt is made to have it particularly steamtight because of its being used only when maneuvering or standing by.

A reliable method of furnishing water at the proper pressure to the glands is important and easily arranged for in land installations by pumping all or a portion of the condensate to an overflow at the required elevation, or providing a tank with a float valve at this location.

For merchant ships there is usually head room enough, and the condensate is usually elevated a sufficient height above the machinery to furnish the required pressure, as shown in Fig. 24, and the system is quite automatic, requiring no adjustment.

On vessels where no head room is available it is customary, as shown in Fig. 25, to provide for taking a small quantity of water from the feed line in which an orifice is provided, which will pass but little more than

that required for the sealing of the glands. This water is carried to the gland system and there is provided a relief valve, by-passing the water not required back to the feed tank, thus maintaining the proper water pressure on the gland system. The gland casings may be removed for cleaning or inspection without otherwise dismantling the turbine.



FIGS. 24 AND 25.

With this combined labyrinth and water gland, provision must be made to turn sealing steam on and gland water off to the gland when reducing to below half-speed, and similarly to turn the steam off and the water on when accelerating above this speed. For merchant ships it is entirely satisfactory to provide two valves, one for the water, the other for the steam, located adjacent to the maneuvering valve, which may be readily manipulated by the man handling the valve. With war ships, however, which require to maneuver more frequently, an automatic valve may be provided, operated in conjunction with the turbine governor shown in Fig. 20. The turbine governor is provided with two springs and its travel is divided into two zones. The inner travel of the governor weights is opposed by the light spring only, the governor weights being able to compress this spring at the speed at which the gland should change over from steam sealed to water sealed. The heavy spring comes into engagement at this speed, but the governor weights are unable to compress the two springs together until the turbine reaches the speed corresponding to which the governor should regulate the steam supply to the turbine.

Motion of the governor through the first part of its travel will, by means of the relay which it operates, either relieve or impose pressure in the chamber "X" of the gland control valve, Fig. 26, causing motion of the valve to either apply water or steam, or *vice versa,* according to whether the turbine is accelerating or retarding.

Couplings.—An important detail is the turbine coupling. To permit free motion of the floating pinion frame there is a flexible shaft provided between the turbine and pinion which provides the necessary element of flexibility. The turbine and gear must be capable of axial motion relative to one another. A coupling of the type shown in Fig. 27 is employed. Means are provided for disassembling either turbine or gear without reference to the other, and proper provision is made for lubricating the driving pins.

Electric Drive.—There is much difference of opinion concerning the relative merits of gear and electric motor drive. Discussion of this important matter is refrained from here in view of certain battleship installa-

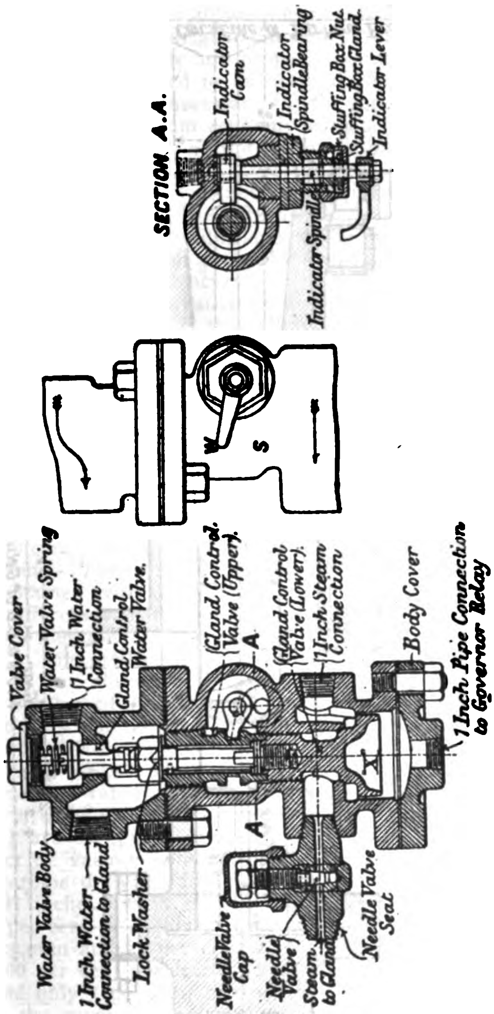


FIG. 26.

tions now being carried out. Unquestionably these installations will be entirely successful, and will fulfil all expectations, but whether the added complication, increased cost and weight of machinery, together with increased steam consumption per propeller horsepower, are warranted, can be shown only by the development of the future.

One system of electric motor drive suitable for the merchant marine has been proposed which has promise of economy greater than that hitherto attained. The proposition is to provide a number of relatively small Diesel engines whose cylinders shall not exceed a diameter of,

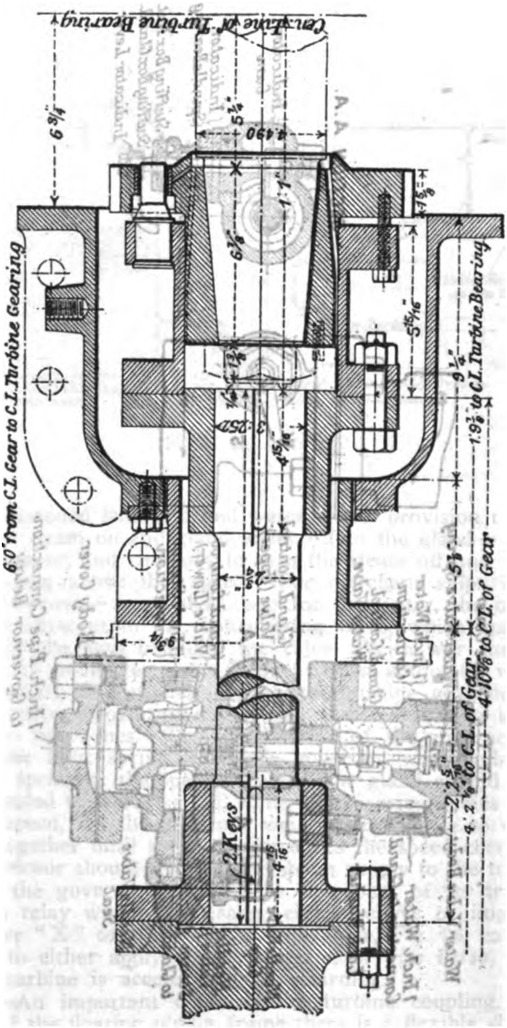


FIG. 27.

say, 10 inches, or within the limits where absolute reliability may be expected. These would be coupled to dynamos which in turn would operate a motor driving the propeller. The locations of the engines may be where desired without reference to the propelling motor.

CAUSES OF IMPAIRED TURBINE ECONOMY.

By J. Y. Dahlstrand.

In addition to its many other advantages the steam turbine has the characteristic of maintaining its original efficiency for a considerable length of time if operated intelligently and properly maintained. In the majority of cases it can also be restored to its original efficiency with a small expenditure compared with that necessary for the same purpose on an engine.

The causes of impaired steam economy with a turbine or turbo-unit in many cases lie entirely outside of the turbine itself. In certain instances, however, the causes are found to develop right in the turbine. When steam economy is impaired it does not necessarily follow that the thermo-dynamic efficiency of the turbine affected is decreased. In some cases the thermo-dynamic efficiency might actually be increased. In the following article the steam economy of the turbine only will be considered, rather than the thermo-dynamic efficiency.

Probably the most common cause of decrease in steam economy, particularly with a small turbine, is the falling off of vacuum in the exhaust chamber. Furthermore, it generally results in the most serious loss.

The effect of vacuum on the thermo-dynamic performance of the steam turbine is well recognized. Various papers and books have been published with charts giving the percentage of steam saved for each inch of vacuum. As a matter of fact, that percentage varies greatly with varying steam pressure and varies somewhat with different types of turbines, with capacities, with operating speeds, with quality of steam, and, last but not least, with the percentage of the designed load developed.

The charts are of two different and distinct types. One shows the difference in steam consumption obtained with turbines designed for different vacuums. The other type illustrates the difference in steam performance obtained with a turbine designed for a certain vacuum but operated at a different vacuum. Both types are subject to variations, due to all the causes mentioned. Naturally the latter is the one which should be considered in this article.

Certain curves are shown in Fig. 1 made by the writer from actual test data on a number of 1000 kw. Kerr turbines designed for 150 pounds steam pressure (dry and saturated steam) and operating at 3600 revolutions per minute, with varying vacua (27 inches to 29 inches, or 68.58 cm. to 73.66 cm.) and loads. These curves may be used with fairly good accuracy for turbines rated at 500 kw. to about 3000 kw. Proper correction must of course be made for steam pressure, if this is not 150 pounds.

These curves are of special interest on account of the fact that they show the effect of vacuum when operating at partial loads. It will be noted that a turbine designed for 28 inches (71.12 cm.) of vacuum, and operating at 27 inches (68.58 cm.), with 25 per cent load, will have a steam rate 12 per cent in excess of that which it would have if operating at 28 inches vacuum and 25 per cent load. If, on the other hand, it was operating at 100 per cent per load and 27 inches vacuum, the steam rate will be impaired only 7 per cent compared with what it would have been if operated at the rated vacuum. The explanation of this lies, as will readily be understood, in the fact that when a turbine is operated at 25 per cent load the steam is throttled before entering the first-stage nozzles to a pressure very much lower than 150 pounds, and the number of heat units constituting the difference in available energy between 27 inches and 28 inches vacuum becomes a much higher percentage of the total energy available for use in the turbine.

Consulting the entropy heat diagram for a verification of the correction figure, 7 per cent, between 28 inches and 27 inches vacuum, it will be found that from 150 pounds to 28 inches vacuum there is 323 B.t.u. available, excluding the effect of reheating and considering straight adiabatic expansion. From 150 pounds to 27 inches vacuum there will be found 301 B.t.u. on the same basis. According to this there is 7½ per cent more heat available with 28 inches vacuum than with 27 inches vacuum. The correction figure for the latter water rate on this basis should be 7½ per cent.

There are certain factors which have a tendency to increase this correction figure. They are: First, the windage losses, which increase with the density of the steam; second, the fact that the nozzle passages in the last stages are too large for the lower vacuum, a circumstance equivalent to operating the last stages at a partial load. These circumstances, however, are generally more than outweighed by another fact.

As was mentioned before, an increase in steam consumption does not necessarily mean that the thermo-dynamic efficiency is decreased. On the contrary, it is sometimes actually increased. This happens to be true to some extent with nearly all commercial turbines of small and medium size. The thermo-dynamic efficiency is largely dependent on the "velocity ratio," or the relation between blade and steam velocities. For commercial reasons the turbine frames are actually in many cases somewhat smaller than those which would give the very best efficiency.

It might be stated here that future tendencies in steam-turbine building will probably follow somewhat different lines from recent practice. Instead of rating a turbine frame at its maximum capacity, without regard for a small loss in efficiency, indications are that the future policy will be to get the maximum efficiency even if at higher cost.

As has already been stated, low vacuum is far from uncommon, particularly in small power plants. It is not unusual to find that a turbine designed for 28 inches (71.12 cm.) of vacuum is actually operating at 25 inches (63.5 cm.) vacuum for months, owing to leaks in the exhaust line or some similar cause, with a loss of no less than 16 per cent in steam consumption. The same percentage loss will be found in coal consumption if the low vacuum is due to air leaks, as this does not lighten the work of the auxiliaries, but rather increases it.

The falling off of vacuum in the exhaust chamber of the steam turbine may be due to a number of different circumstances. The foremost of these are air leaks. These may develop in the exhaust piping between the turbine and condenser, in the condenser itself, or in the turbine glands. Air may also enter through the piston-rod and valve-stem packing on condensate pumps. In addition, air may enter the condenser through the circulating water.

The exhaust-end gland of the steam turbine is generally sealed with water or steam, the simplest form of this gland for condensing service being shown in Fig. 3. This gland is commonly used for impulse turbines. It has three carbon rings with a high-pressure connection. The valve throttling the high-pressure steam is opened sufficiently so that the pressure at the point A is high enough to keep the air from leaking in, and at the same time not high enough to cause any excessive leakage along the shaft towards the outside of the turbine. If too great an amount of high-pressure steam is necessary to seal the gland, it is evident that the packing needs to be replaced. Assurance that there is no leakage of air into the gland may be had by adjusting the seal valve so that a very slight mist of steam escapes along the shaft.

In certain cases with low-pressure turbines, where the steam supply comes from the exhaust of a non-condensing engine, steam leaks some-

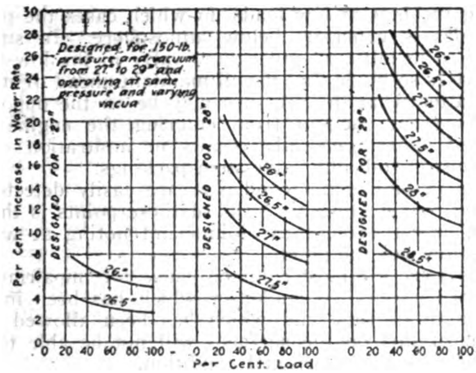
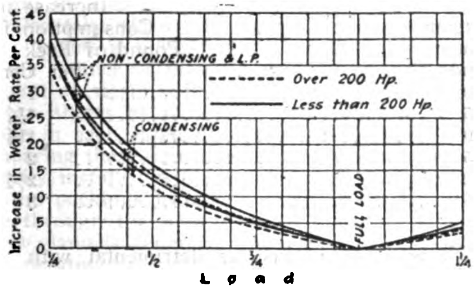


Fig. 1.



Impaired Turbine Economy.—Fig. 2.

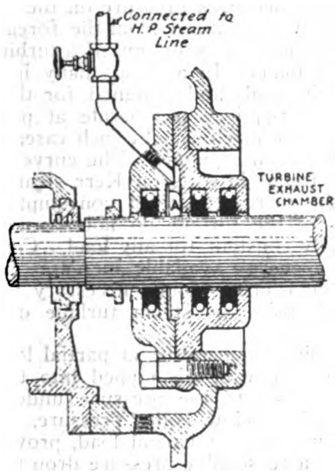


Fig. 3.

times develop in the turbine steam line. These are especially common when operating turbines at low loads, in which cases the pressure on the first-stage nozzles is generally below atmosphere. In such cases it is often advisable to install a so-called flow valve in the low-pressure steam line. This valve is so made that it will maintain ahead of itself a constant pressure, no matter what the pressure may be on the opposite side of it. It will hence prevent vacuum from entering the engine exhaust pipe, which, as stated above, generally results in infiltration of air through pipe joints or piston-rod and valve-stem packings.

The air leaks in piping and condenser are easily detected. The most common method of detecting air leaks at these points is that of bringing the flame of a candle around the joints and noting at what points the flame is drawn towards the point.

While air leaks are the most common cause for impairment in vacuum, there are others as well. A condenser which has been in service for a considerable length of time and which has been allowed to accumulate a large amount of dirt on the surfaces will not be able to maintain the vacuum it could give in its original condition.

Initial Steam Pressure. Pounds.	Increase in Steam Consumption for Each Pound of Back Pressure. Per Cent.
200	1
175	1¼
150	1½
125	2
100	2½
75	3

Excessive back pressure is just as detrimental with non-condensing machines as impaired vacuum with condensing units. This is not so common as impaired vacuum, but may be caused through carelessness on the part of operating engineers in leaving exhaust valves partly closed, etc. The effect of increased back pressure on the steam economy of non-condensing turbines may be realized from the foregoing table.

Next in its effect on the steam economy of a turbine comes the operation of turbines at partial loads. In a great many instances steam turbines are ordered for certain loads and designed for these loads. Later on it develops that they are required to operate at points very much below the full-load rating of the machine. In such cases there generally results a considerable loss in steam economy. The curves shown in Fig. 2 illustrate these losses as found in tests on Kerr steam turbines

The reason for the increased steam consumption at partial loads is obvious. The losses in the turbine do not change in proportion to the load, but remain nearly constant for any load. Consequently the relation of these losses to the energy available increases for partial loads. The throttling of the steam which is found necessary for partial loads, unless hand-valves are used, robs the steam turbine of some of the energy otherwise available.

The fact that a turbine is operating at partial load may be seen by observation of a steam gage which is tapped into the ring chamber ahead of the first-stage nozzles. If the pressure under normal operation at this point is considerably below boiler pressure, it is an indication that the turbine is operating under a partial load, provided, of course, that it was designed for only a reasonable pressure drop through the valve. The power developed with any type of turbine is very nearly proportional to the pressure at that point.

Turbines should be made to operate normally under the conditions for which they are designed. If it is necessary that they should carry overloads, overload valves—either automatic or hand-operated—should be used.

Up to this point only those circumstances which arise through steam conditions and from outside sources have been considered. Certain mechanical conditions inside of the turbine may also cause a thermodynamic loss in the steam turbine. Foremost among them is internal leakage. This is caused by the wearing out of the packing rings or bushings or the excessive clearance of labyrinth packing between the various stages in the steam turbine.

Internal leakage is, of course, most liable to develop where the pressure differences are very great. Therefore, in certain makes of turbines the high-pressure steam space is separated from the vacuum chamber with an internal gland. In cases of this kind it is always advisable to watch this gland carefully, as leaks may readily develop through it, causing a considerable loss in steam economy. Internal leakage is particularly detrimental in the high-pressure stages of a machine on account of the low specific volume of the steam in these stages, which results in a large discharge of steam through a small opening.

Leakage may also take place between the stages along the horizontal joints of the diaphragms in a multi-stage turbine. The steam, and moisture may erode the surface of these diaphragms to such an extent that a considerable amount of steam will pass through from stage to stage.

Leakage between stages as well as internal leakage may be detected by placing a gage in each stage of the turbine to give the pressure in that stage. Comparing these figures with the pressures which the turbine had when originally installed will give a good indication as to the condition in which the packings installed in the turbine are.

Steam leakage through the high-pressure gland or through the leakage piping on this gland is generally negligible. It is usually sufficiently annoying to the operating engineer, however, to lead him to try to overcome this condition. Generally, losses due to this cause are greatly exaggerated by operating engineers.

Among the causes for impaired steam economy is excessive clearance between stationary and rotary elements. When this condition exists, Rateau, Curtis, and other impulse turbines lose slightly in power. It would be difficult to formulate any definite rule as to the degree in which steam economy is influenced by clearance, as this varies with capacity of machine, nozzle, and blade angles, etc. It is advisable, however, to bear it in mind if it becomes necessary to manipulate the thrust bearings. In general, it may be said that the rotary element should not be farther away from the stationary element than is necessary for the mechanical safety of the machine. Each case should be considered separately, however, so turbine users should consult with the manufacturers of their machines if information regarding axial clearances is desired.

With certain makes of turbines, particularly single-wheel impulse turbines, where the total pressure drop is utilized in one single expansion, considerable trouble has been experienced with erosion. This has been the case particularly with turbines in which the relative steam velocity was high—in other words, turbines having small wheel diameters or low speeds and operating condensing. Some difficulties have been experienced even with non-condensing turbines as well, particularly where the steam has been wet. If superheated steam is used, erosion is less common. Blade materials now being used in various types of steam turbines utilizing high steam velocities are being improved so as better to resist the erosive action of the steam. Erosion is absolutely unknown with other types of turbines on account of the low steam velocities employed. Through erosion of the blade edges the axial clearance between the

rotary and stationary elements is increased, which tends to cut down the power developed. Through the fact that these edges are made dull less perfect steam action is attained, which in turn causes a loss in power and efficiency.

The deposits of foreign substances in the turbine blades, blocking or partly blocking the passages, might be mentioned as another cause of impaired steam economy in turbines.

This trouble usually occurs when a turbine is being fed by one or more boilers which are giving about their maximum capacity. It is more serious with some types of boilers than others. In both fire-tube and water-tube boilers there is such active circulation of water that the mud and foreign material are prevented from settling in the boiler and are mechanically lifted to the top and carried through the steam lines.

In some steel-mill districts the deposit consists principally of a muddy material which is bound together by chemicals existing in the water. At the velocity at which steam enters the blades any slight material carried in the steam will be impelled against the blades with such a velocity that it will form a very compact deposit. In a great many instances the boiler-feed water supply contains large portions of vegetable matter, as well as boiler scale-forming materials. Where this is the case the combination produces a tough, rubber-like deposit which rapidly fills the turbine blades and causes serious trouble.

Owing to the immense quantity of steam passing through large turbines a rapid increase of deposits may result even though the amount of material mechanically carried over by the boiler is very small per unit of power. It is, therefore, frequently astonishing to note the accumulated deposits when the steam is apparently quite pure and clean. Sometimes turbine blades become so clogged that sufficient power cannot be produced and the turbine has to be dismantled and cleaned.

Steam that is very wet, when made from water containing scale-forming material, such as magnesia and other salts, will invariably deposit a scale-like substance over the turbine blades. This is somewhat distinct from the deposits mentioned previously. There are other deposits which are more local in character, such as, for instance, deposits which occur in paper-mill districts, where a considerable amount of pulp is discharged in a finely divided state into rivers adjacent. The result is that this finds its way into some neighboring power plant where, owing to its fine and light nature, it is carried over by the steam and very rapidly blocks up the turbine blades. Several cases have come to the writer's attention where this occurred, the turbine having to be dismantled at regular intervals and the blades cleaned out three or four times a year.

There are various means of combating the clogging of blades from the above-mentioned causes which are effective in different degrees. One which is frequently used, particularly for non-condensing turbines, is the placing of a lubricator in the turbine steam line. The oil vapor in the steam lubricates the blade surfaces and prevents the substances from becoming attached. Even for condensing turbines this method is frequently used, although it would appear not to be entirely advisable in installations where surface condensers are used, on account of oil deposits which would be found on the tubes. The writer knows of two instances where this method is used with turbines rated at 10,000 kw.

Undoubtedly the most effective means of preventing this trouble is that of installing a large receiver in the steam line. The steam velocity will be decreased greatly while passing through this receiver, and the foreign substances will be deposited on the walls. In addition to the receiver there should be installed near the turbine throttle a steam separator which will serve the double purpose of removing from the steam any remaining foreign substances and extracting a great deal of

moisture from the steam. It might in some cases be advisable to combine these two apparatus and install a so-called receiver-separator, in which case the foreign materials will be deposited on the baffles.—"Electrical World."

THE "DEX" OIL COOLER.

In view of the developments which are taking place and the continued increase in the size of prime movers, the problem of efficiently cooling the large quantities of oil required for the bearings of turbines, thrust bearings and gear boxes—like many of the details in a large power installation—plays an important part in the smooth running of the station, and has compelled designers and those responsible for the running and maintenance of such installations to investigate the subject in some detail.

We have received particulars of the "Dex" oil cooler (Harrison's patent) which is being manufactured by Messrs. W. H. De Ritter and Co., Limited, of Limehouse, London, E. This cooler is of a new type and, being compact and very accessible for cleaning, has advantages over many of the older patterns of tubular coolers. The leading features of the "Dex" cooler are shown in Figs. 1 and 2; they consist in the use of flat plates cast with ribs on both sides, assembled alternatively with plain rolled plates, the nest of plates being held together by bolts connecting heavy cast-iron headers on which are arranged the inlet and outlet connections for the oil and the cooling water. The ribbed plates are die cast and then ground on both faces. This work must be carefully carried out to insure that when they are assembled with the plain plates the joints are sound. This is essential, not only to avoid short paths for the fluids, but to prevent leakage from the oil passage to the water passage, or *vice versa*. When it is desirable to clean or examine the plates, it is only necessary to withdraw the bolts, when each plate can be readily removed and brushed over.

The oil and water are circulated through the rectangular passages and in reverse direction in the alternate layers, and after circulation are delivered to their respective pockets, as indicated by the arrows on Figs. 1 and 2.

In Fig. 2, all the oil and water passages are in parallel, but it will be readily seen that by the introduction of baffles in the pockets the parallel system of working may be changed to the series system with two, three or more flows, thus increasing the path to be traversed by the oil to be cooled.

The shallow rectangular passages for the fluids give a high heat transmission efficiency without undue friction, and from tests which have been carried out by the manufacturers it was found that a transmission factor of 270 B.t.u. per 1 degree F. per square foot was obtained for a drop of 15 pounds pressure between the inlet and outlet connections. It appeared, moreover, that this transmission factor varied in direct proportion to the velocity of the oil and that the velocity was nearly proportional to the pressure.

The makers of this cooler are prepared to supply them for the services already mentioned, and also for cases where it is required to transmit heat between two substances one of which is a gas or vapor.—"Engineering."

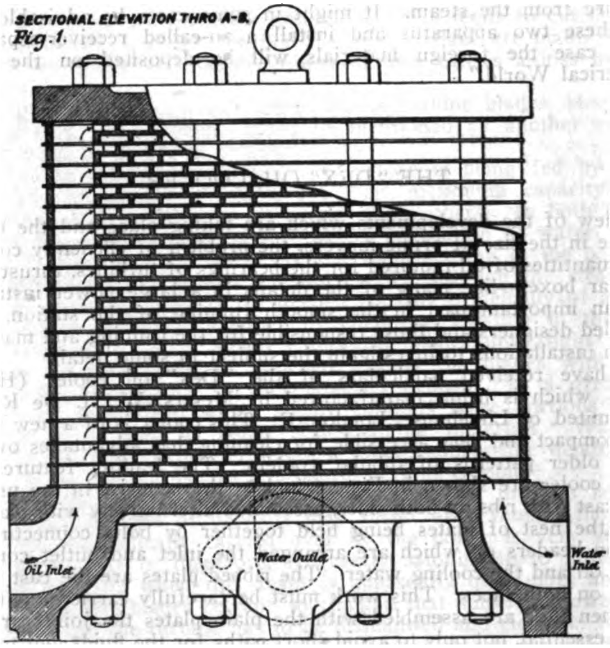
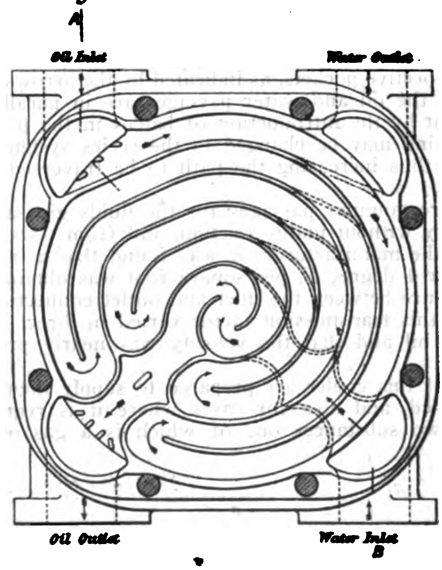

SECTIONAL ELEVATION THRO A-B.
Fig. 1.
Water Outlet
Oil Inlet
Water Inlet

Fig. 2.


A
Oil Inlet
Water Outlet
Oil Outlet
Water Inlet
B

SOLDER, ITS USE AND ABUSE.*

By MILTON L. LISSBERGER,† New York, N. Y.

Solder is a mechanical mixture of tin and lead, a fact which is susceptible of very simple demonstration. A bar of solder of a grade even as low as 30 per cent tin and 70 per cent lead, passed through a buffing machine, will show a surface practically identical with that of a bar of second-quality or reclaimed tin. The buffings, on chemical analysis, will prove to be almost pure lead.

According to the best practice, solder is made in the following manner. Virgin pig lead is first melted, and when it is thoroughly liquefied, virgin pig tin is added, together with a small amount of flux; the latter is for the purpose of bringing to the surface the so-called "liver," consisting of impurities that may have remained in either the lead or the tin as a result of incomplete refining. The combined material, when completely liquid is thoroughly stirred for some hours, and is then cast into small pigs. Just before casting, and continuously during this operation, the molten metal yields dross, consisting largely of the oxides of lead and tin; this should be carefully skimmed off.

After the pigs have cooled, they are taken to a smaller kettle, remelted, and cast into the desired shape for use; or if wires, ribbons, etc., are to be made, the solder is cast into slugs suitable for extrusion and rolling. During this second operation, the skimming of dross should be even more carefully done than at first.

Hand mixing has proved to be the only reliable method for the production of the best quality of solder, irrespective of its percentages of lead and tin. The best quality of solder is not necessarily that which contains the highest percentage of tin, but rather is that composition which performs best on the required piece of work. In order to produce a thorough mechanical mixture, it is necessary to stir for a long period; experience has shown that to perform this operation satisfactorily takes from 5 hours to 6 hours, irrespective of the quantity of material being mixed, and also irrespective of the proportion of tin in the mixture, whether 60 per cent, or as low as 30 per cent.

Throughout the casting process, what occurs is that the lead solidifies in skeleton crystals until the remaining liquid has the eutectic composition, when it freezes at a constant temperature as a mechanical mixture of tin and lead containing some tin in solid solution. It is remarkable how many shapes these skeletons take. The seeming explanation of this variation is the presence of other metals than tin and lead, in very small proportion, or even traces.

With the idea of conserving tin, solders should be separated into two classes:

(1) That which is used strictly for soldering, that is, joining and holding together two pieces of metal.

(2) That which is used primarily for the filling of an interlocked joint, so as to prevent the escape of the contents of a container. It is these filling metals that offer the greatest opportunity for the conservation of tin. It is only necessary that the metal shall flow into the seam, and solidify into an impenetrable mass.

The greatest abuse of solder occurs in the use of high-tin mixtures for filling metals. A mixture of 25 per cent tin and 75 per cent lead, worked at the right temperature and with proper fluxing, is high enough in tin

*Paper read before the American Institute of Mining Engineers (slightly abbreviated).

†President, Marks Lissberger and Son, Inc.

for any filling purpose, as has been demonstrated in the practice of the oil canners, notably the Standard Oil Company.

The filling operation is usually conducted by machinery, but the users have frequently not realized that the baths are considerably richer in tin at the top, through which layer the container is being dragged, than the solder that is put into the baths. When the 40 : 60 solder, most commonly used on automatic can-making machines has not worked entirely satisfactorily, it has often been found that the addition of 1 inch or 2 inches to the depth of the bath has made the solder work very much better. Hence, one of the best means of conserving tin in can-making solder is to deepen all baths, whether on line machinery or for hand dipping, thus permitting the use of a lower-grade solder.

The fact that solder dross contains a higher percentage of tin than the original solder has usually been explained on the assumption that tin oxidizes more rapidly than lead. The probable explanation is that in solder baths the lead is gradually working toward the bottom and the tin to the top, where it is exposed to the oxygen of the air, thus the oxide of solder is richer in tin than the original solder.

The overheating of solder is not only detrimental to the work, but also causes some, though not a very great, waste of tin through the excessive production of oxide. While this oxidation may be a source of considerable expense to the package manufacturer, it is not actually a very serious loss of tin because the reclaiming of these drosses, or oxides, has been so perfected that very little of the original metallic contents is lost. In these days, however, when every ounce of tin should be conserved, both to insure a sufficient supply for the most essential work, and to save the useless transportation of a material which comes such long distances by boat, overheating should be avoided, and all baths should be covered with a protecting material such as sal ammoniac, oil, charcoal, or ash.

In this connection, it should be emphasized that every particle of solder oxide should be preserved, and sent to the reclaimer. A teaspoonful of solder dross contains enough solder to make a 5-gallon can or 100 No. 1 cans. In many plants, even those of some of our largest consumers of solder, this dross is not collected and saved with sufficient care. It is seldom that a thorough cleaning and gathering together of the oxides takes place more than once a week.

Fire will purify these reclaimed materials, when properly refined, and in purity they will compare favorably with the virgin materials. However, too little attention has been paid to the proper refining of these scrap metals. Usually they have simply been put into a kettle, melted down, and then brought up or down to the required composition. This is not sufficient. Reclaimed metals are never equal to virgin metals, no matter how much refining they undergo; nevertheless, for certain classes of work they are economical and efficient. The repeated use of metal affects its physical permanency; yet the margin of safety in the use of solder is so large, and the length of time that solder is required to remain on the container is so comparatively short, that any lack of permanency can usually be safely disregarded.

After many years of experience, we have developed the following method of manufacture. The lead is first melted at a temperature which does not cause too rapid fusion. After the dross has had a chance to rise to the surface, it is carefully skimmed off before the required amount of tin is added, and slowly reduced to the liquid state. From the moment the tin is added, the solder is stirred by hand for 3 hours to 4 hours. A scavenger is then added and thoroughly worked for another 3 hours to 4 hours; the resulting dross is again skimmed, and the solder is cast into pigs of approximately 80 pounds each. These pigs are then re-melted in smaller kettles at a temperature which just causes free fluidity, and

the solder is then cast into the desired shapes. During the entire operation of final casting the caster stirs every time he takes a ladleful from the pot.

This work could be done much more rapidly at higher temperatures, and much more economically by the use of mechanical mixers, but the resulting solder would not be so thoroughly mixed, nor would it be so fluid. Mechanical mixing has a tendency to drive the oxide and dross back into the metals, thus diminishing the holding power of the solder. The scavenger must be chosen with great care, and the amount must be very accurately gaged; otherwise the scavenger becomes a constituent of the finished product, and, instead of being beneficial, is a detriment. We have found that in the grades of solder containing 46 per cent and less of tin, the addition of ½ per cent to ¾ per cent of the best grades of antimony increases the fluidity and holding strength of the solder for working tin plate.

Next to its use for containers, the largest consumption of solder has been on gasolene-motor radiators. The hand work on these radiators requires merely a free-flowing clean solder, but on the dipping work, where most of the solder is used, the greatest abuse has been practiced. As these radiators are composed of copper, low brass, or ordinary brass, no antimony whatever should be added to the solder used for this purpose. Also the affinity of tin and lead for zinc and copper will draw both of these metals from the radiators into the baths, and as both copper and zinc make solder sluggish, it does not take long (unless proper methods are employed for cleansing the baths) for the solder to become deteriorated.

These baths can be thoroughly cleaned by a mixture of rosin and sulphur, but as this operation produces very disagreeable black smoke throughout the plant, some method should be devised for disposing of it. When sulphur is used for removing zinc and copper, a sufficiently high temperature should be employed to insure the complete combustion of the sulphur. The baths should then be allowed to settle for at least half an hour after such heating, and the top carefully skimmed to remove any sulphides present. It is important to note that the presence of any non-metallic substance is injurious to solder, whether it has been added as a scavenger or is liberated from the original metals.

The question is frequently asked, what is the strongest solder that can be made. Numerous experiments have been made, but the results are confusing. Tests of tensile strength, based upon wires and cast bars, indicate that the higher the tin, up to 75 per cent tin, 25 per cent lead, the greater the breaking strength; in the case of two pieces of tin plate soldered together, the maximum strength is given by a solder containing around 42 per cent tin. Other tests were made on square 5-gallon cans, completely filled with water and then capped; when dropped from a height of about 100 feet, the cans soldered with 46 per cent tin, 54 per cent lead, in no case broke at the seams, although the tin plate was ruptured. This was the only mixture that gave this result. Cans soldered with 47 per cent or more of tin, 53 per cent or less of lead, and with 45 per cent or less of tin, 55 per cent or more of lead, occasionally ruptured at the seams. These experiments were made most carefully and were afterward confirmed by subjecting the cans to air pressure.

I am thus inclined to believe that, in round figures, 46 per cent tin, 54 per cent lead, is the strongest mixture that can be used for general soldering purposes, particularly if ¼ per cent to ½ per cent of antimony be added to the mixture. The Bureau of Standards, with the approval of the War Industries Board, suggests that the highest grade of solder permitted should be 45 per cent tin, 55 per cent lead. For mechanical soldering, 40 per cent should be the highest tin ratio, and for most bath work it has been demonstrated that tin from 35 per cent to 38 per cent, according to the nature of the work, will give ample satisfaction, provided the solder is made properly.—"Engineering."

ELECTRIC HEATED INDUSTRIAL FURNACES

By George J. Kirkgasser

War conditions have greatly increased the use of the electric furnace, not only in work where it was already used, but where it had hardly been given more than a little consideration. Because of the comparatively short time that the electric furnace has been commercially used, methods of operation have not been so well standardized, but with the number of men now familiar with them, there will be an interchange of ideas that will help toward standardization.

Where temperatures high enough to reduce steel to a molten condition are required the electric-arc furnace is employed, while for subsequent heat treatments, resistor-type furnaces are used, for in the latter process great accuracy and exactness of temperature control must be maintained.

The electric furnace is a metallurgical appliance in which any desired temperature up to the point of fusion of the best refractory materials obtainable can be attained and perfectly controlled. At these high temperatures chemical reactions take place much more rapidly than in other processes, and the most refractory metals and alloys can be reduced to fluidity.

An electric-arc furnace for steel making usually consists of a steel tank lined inside with refractory materials and fitted with working doors, spouts and tilting arrangements for teeming; regulatable carbon or graphite electrodes of a suitable section are inserted through the roof or walls. A high-tension electrical supply is transformed down to a low pressure for operating the furnace.

Arc furnaces are of two principal kinds: one known as the arc-radiation type, in which heat is radiated from the arc; and the arc-resistance or arc-conduction type, in which there is radiation of heat from the arc, and heat also generated as a result of the resistance due to the flow of the current in the furnace.

PRODUCTION OF ELECTRIC STEEL.

What are said to be the first patents on the electric arc steel furnace were recorded about 40 years ago, an Englishman, William Siemens, being the inventor. As with other patents, the patentee could not by any stretch of imagination realize to what extent the electric furnace would be used in the steel industry. The world's output of electric steel has multiplied more than tenfold since 1914. In that year there were about 30 electric furnaces in the United States and Canada, while at the beginning of 1918 there were 300 such furnaces in use in these countries with an annual output of about 2,000,000 tons.

The Illinois Steel Co. of South Chicago has the largest output of electric steel in the country, employing three 25-ton and two 15-ton furnaces of the Heroult type. The first, a 15-ton furnace, was installed in 1909. The products of this first furnace, steel rails, were laid on 14 railroad lines and the results showed that the electric steel was more ductile at low temperatures than either the openhearth or the Bessemer steel. Subsequent tests showed this also. Now the capacity of the plant is 16,000 to 17,000 tons of electric steel per month.

The electric furnaces manufacturing steel in England are now producing about 40 times the pre-war output, and it is now possible to do without the large imports of Swedish iron and steel which were formerly considered necessary to maintain the output of high-class products. Furthermore, the approaching exhaustion of the high-grade ores of

Cumberland may be looked upon with equanimity, as electricity makes possible the use of the inferior ores of Cleveland and the Midlands to manufacture steel of great purity.

The electric furnace has not only proved that it offers many advantages in steel making over the older processes, but has also shown that in making many classes of steel it is an absolute necessity.

Electric steel is free from dissolved gases which is due in a measure to the perfect control over the temperature of the steel by the operator. By raising the temperature until the steel is in an exceedingly fluid state, the gases are permitted to escape, after which the temperature can be regulated for casting so that the desired grain can be produced in the setting of the ingot.

In practically all cases the electric furnace is used not for the manufacture of steel directly from cold metal, but more as an electric refining process. Making steel directly from the cold metal requires a large consumption of current which makes the cost prohibitive. Usually it is used to finish metal previously made by the Bessemer or openhearth process.

Very often in steel plants, the openhearth furnace is used for preliminary melting and an electric furnace for deoxidizing and desulphurizing, and for making any other changes needed to secure a steel according to analysis. In what is known as the "Triplex" system, the Bessemer converter is used for decarbonizing and desiliconizing the metal; the openhearth furnace is then used for dephosphorizing it and the electric furnace completes the work of deoxidizing and desulphurizing.

BASIC OR ACID OPERATION.

The basic process of furnace operation is the more costly because of the present high cost of refractories. The use of basic slag serves to remove impurities from the metal, just as in the basic openhearth steel furnace. Phosphorus, oxygen and sulphur may be removed, which permits using relatively cheap raw materials. With acid operation no phosphorus or sulphur is removed and more care is needed to hold the elements between desired limits.

In lining basic furnaces, magnesite or dolomite is used up to about the slag line and above this silica, magnesia, and chrome brick is provided; while for the roof, silica is used.

For acid operation, silica is used for the entire lining. Acid operation has been used to a great extent because of the abundance of shell turnings, steel clippings, and punchings from steel shipyards, etc. It is a little quicker than the basic, as it is not necessary nor possible to refine the charge.

In Canada electric pig iron is made from steel scrap, the scrap being used in this way rather than being turned into refined steel, because of the shortage of low phosphorus pig iron. About 500 kw. hours of power per ton is average current consumption in this process.

Dr. John A. Matthews, president of the Halcomb Steel Co., Syracuse, N. Y., recently said:

"The electric furnace requires for successful operation less labor per ton than is necessary for crucible steel, but it calls for metallurgical skill in order to produce the best results. Easy oxidizing metals like vanadium, chromium and manganese, are readily handled and less of them required to give the content desired in the steel. Sulphur and phosphorus can be readily eliminated and the yield of sound steel increased.

"With the electric furnace, alloy steels are made in the furnace itself rather than in the ladle, and in this way there is better opportunity for increasing the solution, diffusion and homogeneity in the product. All

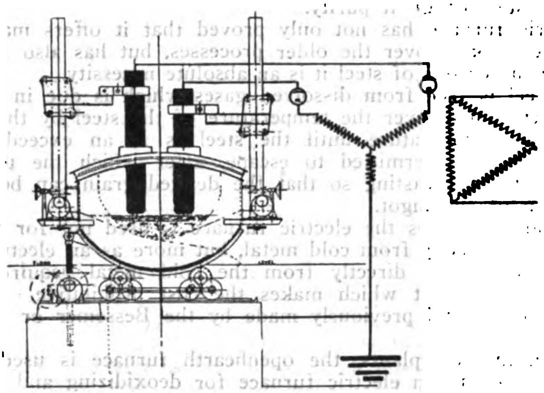
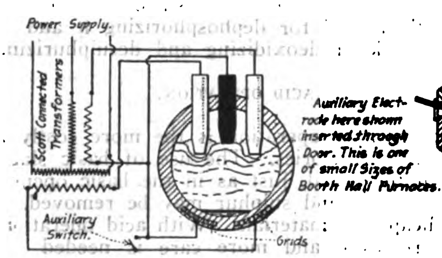
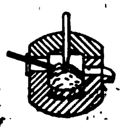


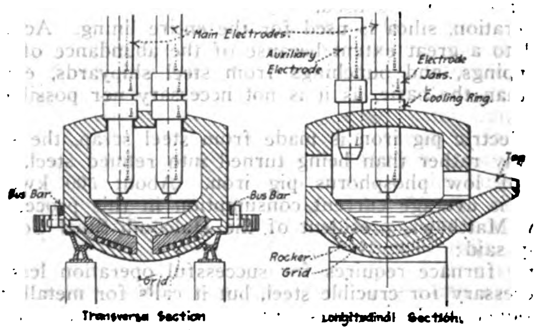
DIAGRAM SHOWING ELECTRICAL CONNECTIONS AND CURRENTS SET UP IN THE MOLTEN METAL IN A GREAVES-ETCHELL FURNACE



ELECTRICAL CONNECTIONS OF BOOTH-HALL FURNACE WITH TWO-PHASE CURRENT SUPPLY



AUXILIARY ELECTRODE INSERTED THROUGH DOOR



BOOTH-HALL FURNACE—TWO-PHASE CURRENT SUPPLY

of these things make for high quality, and quality is the first consideration. In addition to this, the electric furnace performs an economic function because of its adaptability for handling and recovering alloy values contained in scrap. This was of especial importance during the war, when alloys had to be conserved. The alloy content of chromium, manganese and vanadium in scrap used in the open-hearth steel is only recovered to a very small extent, and not only is the alloy value lost but the oxides formed are frequently a source of trouble in the final product."

The superiority of electric steel castings over those made by any other process has been conclusively proved by the extremely severe conditions to which they have been subjected under specialized war conditions. The castings produced can be finished off dead mild and a one-inch square bar can be bent as cast, no expensive annealing operations being necessary. The carbon contents usually vary between 0.2 per cent and 0.25 per cent, while the sulphur and phosphorus are each reduced below 0.015 per cent and the manganese and silicon are adjusted to suit the work.

The tensile properties of ordinary electric cast steel are:

Maximum tensile strength, 35 tons per square inch.

Yield point, 25 tons per square inch.

Reduction of area, 50 per cent.

Elongation, 30 per cent.

The maximum tensile stress can be readily increased to even 100 tons per square inch by the introduction of special elements such as nickel, chromium and vanadium.

Steel from the electric furnace is finished off under a reducing atmosphere of carbonic oxide and a slag out of which the metallic oxides have been reduced, and is singularly free from blow-holes when properly cast. This is due to the elimination of the gases that molten steel usually holds in solution and the ease with which the steel can be poured into small and intricate castings, flowing smoothly and setting perfectly.

Electric steel castings can be made as cheaply as malleable iron castings, while the former are lighter for the same strength.

HEROULT FURNACES.

Heroult furnaces are of the arc-resistance type, the electrodes for introducing the current being placed above the bath and the current passing from one electrode to the metal and from the metal to the next electrode. The bath is thus heated by radiation from the arc and by the resistance of the steel and slag to the flow of current. The steel shell is lined with a refractory material, the nature of which depends on whether the furnace is to be basic or acid in its operation.

The 25-ton furnace of this type at the Duquesne plant of the Carnegie Steel Co. requires 175 kw. hours per ton of steel and operates basic on hot metal. The three electrodes are of 12-inch graphite and are used up to the extent of about four pounds per ton of steel. Three phase current received at 6,600 volts is transformed to 100 volts at the furnace. Six heats are made per day of 24 hours, and each heat averages about 2½ hours.

GREAVES-ETCHELLS FURNACE.

In principle, the "Greaves-Etchells" Furnace is a combined arc-resistance furnace.

Two phases of a three-phase low-tension supply are connected to their respective upper graphite or carbon electrodes, while the third phase is connected to the bottom of the hearth. The current flowing through the hearth generates a considerable amount of heat immediately below the

liquid in the most efficient manner possible, while the electric arcs arranged over the bath maintain the slag and surface at the desired temperature.

The effect of this bottom heating is to cause convection currents in the molten metal, which insure a constant circulation and a uniform product. The outside of the furnace bottom remains cold, little or no heat being lost in this direction.

Charges of high-speed steel and castings have been made from cold scrap in one hour and ten minutes. Present sizes include 1, 2, 3, 5, 6, 10, 12½, 20 and 30-ton furnaces. A half-ton furnace is also made specially for high-speed steel, small steel castings and special tool steel.

BOOTH-HALL FURNACES.

The Booth-Hall furnace is built for use on single, two and three-phase electric power lines, in the following capacities: ¾, 1½, 3, 6, and 10-ton. Power is introduced to the furnace through one, two, or three main vertical electrodes, depending on whether the furnace is single, two, or three-phase. An auxiliary electrode is also introduced and there are cast-steel grids imbedded in the hearth with a refractory facing. The Booth-Hall furnace is therefore of the vertical arc type with conducting hearth.

In starting a heat with a cold charge in the two-phase Booth-Hall electric furnace, a pawl on the auxiliary electrode holder is released, permitting the auxiliary electrode to rest upon the scrap charge. Arcs are then drawn with the two main electrodes, and the metal begins to melt under the arcs. The auxiliary electrode acts as a common neutral, or return, no arcs being drawn and, therefore, no metal melting under the auxiliary electrode. When a molten bath has accumulated upon the hearth, sufficient to make the hearth conductive of electricity, ammeters indicate that current is beginning to flow through the cast-steel grids. The auxiliary electrode is then raised up out of contact with the charge and plugs its own hole in the roof. The furnace then continues to operate with the current entering the furnace through the main vertical electrodes and being carried through the bath and the hearth to the grids, and so out through the grids. The connections from the transformer to the furnace are so arranged that each of the two phases has an independent circuit.

At the West Michigan Steel Foundry, the three-ton furnace has averaged 2 hours 15 minutes per heat and the current required per net ton is about 676. Records of the analysis for the months of August and September, 1918, are given. The operations are all on a basic furnace lining.

The Snyder furnace is also of the arc-resistance type several of which are in operation in the plant of the Haynes Stellite Co. of Kokomo, Ind.

The Rennerfelt furnace is of the arc-radiation type used mostly in 1-ton and 3-ton sizes in various industries as foundries, machine makers, valve makers and some in the non-ferrous industry.

AVERAGE ANALYSIS.

Element.	Per Cent.
Carbon	0.25
Manganese	0.55
Sulphur	0.046
Phosphorus	0.045

AVERAGE PHYSICAL PROPERTIES.

Tensile Strength	63,040 pounds.
Elastic Limit	38,010 pounds.
Elongation	29.9 per cent.
Reduction in Area	41.5 per cent.

ELECTRIC FURNACES IN FOUNDRIES.

Electric furnaces are today in operation in foundries for melting steel scrap, refining steel scrap, refining molten steel and melting gray iron, malleable iron, ferro-manganese, ferro-alloys, copper and copper alloys. In some foundries the electric furnace is used for the melting and refining of steel, where an extra high quality of steel is desired, low in sulphur and phosphorus and meeting definite specifications. The Canadian and United States governments have both installed a considerable number of electric furnaces for this purpose, the chief object being the production of steel low in sulphur. Eventually the application of the electric furnace to the melting of malleable iron will be as important as it is to the steel industry today. Ordinarily, the electric furnace cannot compete commercially with the cupola for melting gray iron. It is useful, however, in manufacturing aeroplane motor cylinders of very high-grade gray iron, owing to the facts that the sulphur content is kept down, that the metal is melted under conditions ideal for the production of iron free from air-holes, and that the iron produced from an electric furnace has the same density that is characteristic of electric steel.

Mr. W. E. Moore, president of the Moore Electric Furnace Co., said at the recent Metal Congress and American Foundrymen's Convention in Milwaukee, that basic-lined furnaces were most suitable for the ordinary steel foundry although at present more acid-lined furnaces are in use because specifications are liberal as to sulphur and phosphorus content. High-grade scrap has been available because of the manufacture of munitions. The acid-lined furnace is considered simpler and cheaper.

Because of the ease of operation of the electric furnace, small foundries can now make high-grade steel castings, replacing iron castings to a great extent. In addition there is a wide field opening up for alloy steel castings of a grade that can be properly made from electric-furnace steel. Such castings can be heat-treated and may in some cases replace drop forgings.

HEAT TREATMENT OF METALS.

Heat treatment, electrically came into its own with a vengeance during the past two years. Crankshafts for Liberty motors, bolts and nuts for airplanes, gears for our army trucks, cast-steel anchor chains for the emergency fleet, gun parts, locomotive axles have all been treated in electric annealing furnaces. One furnace is 193 feet long, used for annealing bright strip steel, the guarantee being that the steel will come cut of the furnace as bright as it went in. The strip is fed through on cars which move continuously through the furnace, the capacity for a day being 150 tons.

Heat treatment includes "hardening," "annealing," and "drawing" or "tempering." The furnace temperatures vary from 1,550 to 1,650 degrees F. The accurate control possible with the electric furnace makes it easy to provide exactly the temperature required, and this is important because if the temperature is too low, the results are not obtained; while if too high, the steel is damaged—"burned."

Electric means of measuring and the pyrometer make it possible to arrive at a standard, and to duplicate the conditions. This eliminates waste and improper results. The heat from the electric furnace is also uniform and the temperature of the metal is raised uniformly.

The resistance electric furnace has the usual firebrick walls and insulating material enclosed in steel frame work. The resistance elements, located entirely within the furnace chamber, are rectangular in cross section and run the entire length of the furnace, or, in the case of circular furnaces, entirely around the furnace. This trough is composed of a carbide of silicon material. The electrodes for leading the current through the furnace walls are of carbon or graphite. The resistance material is laid in the refractory trough mentioned and is composed of broken carbon or graphite. The current flows from electrode to electrode through this carbon resistor, which heats up in the manner of the ordinary carbon-filament lamp, the heat radiating from the carbon and the trough heating the furnace chamber, and, in turn, the material to be heated. The control of the heat input is obtained by means of various voltages from the secondary of a special transformer, a relatively definite number of kilowatts being obtained from each of the various voltages.

Hearth-type annealing furnaces are adapted for small pieces as gears, nuts, bolts, etc. The Baily furnace of this type has a capacity for heating 200 pounds of steel to a temperature of 1,620 degrees F. in one hour with a current consumption of 400 kw. hour per ton of metal heated. The temperature can be held within limits of five degrees variation.

With larger furnaces of the car and pusher types, greater economy of current is obtained.

Automatic operation has attracted the widest attention because of the results obtained. A heat-treating equipment of this kind was referred to in a paper read by Mr. Baily, president of the Electric Furnace Co. of Alliance, Ohio. This consists of two furnaces, each having a hearth 20 feet long and 7 feet wide. In the treatment of 14,000 tons of steel not a single piece of material was rejected due to faulty heat treatment.

These furnaces are automatically controlled by contact-making pyrometers, which, in conjunction with a special electrically-operated valve pulpit, control all the operations of the furnace. The pyrometer couples located in the discharge end of the furnace are dominated by the temperature of the material itself. Therefore when the material at the discharge end has reached the temperature required, the pyrometer couple actuates the pyrometer instrument, which, in turn, through suitable relays, sets in motion the valve pulpit device, which in proper sequence raises the door of the charging furnace, swings the manipulator platform under the door of the furnace, and by means of the pusher pushes in a fresh charge of material at the same time discharging on to the manipulator platform the material located under the pyrometer couple which starts the operation. When this material has become centered on the charging platform, it is removed by the manipulator from the furnace and plunged into the quenching bath as the door closes.

At this period of the operation a time-element device is thrown into circuit, which holds the material being quenched in the bath a definite time, after which the manipulator raises the material out of the quench and places it in front of the charging platform of the second furnace, where it remains at rest until the pyrometer mechanism of the second or drawing furnace puts into motion a similar operation to that just described, on which the heat-treated material is discharged from the second furnace.

Besides the use of electric furnaces in the making and treatment of steel, they are also employed for treating brass castings, silver, copper, phosphor and manganese bronze and other non-ferrous metals. Arc-type

and resistance-type reverberatory furnaces are used. There are a number of furnaces in the experimental stage for this class of work.

For the melting of alloys as nickel, steel and nichrome, the Driver-Harris Co. is successfully using a 2-ton Heroult arc furnace operating at about 2,200 degrees F.

A plant of five electric furnaces have just been put in operation by the Anaconda Copper Co. for the production of ferromanganese from low-grade manganese ore.

In the manufacture of Stellite, an alloy developed especially for cutting tools, the electric furnace seems to meet the requirements. A resistance type was tried and later three Snyder arc-type furnaces installed. Power consumption averages about one-third kilowatt per pound of metal melted, the melting time required being about 30 minutes.

HOSKINS FURNACES.

Small electric furnaces of the type generating heat by means of passing current through a resistor or heating element are used for many purposes, such as ash determinations, enameling (as watch dials in watch factories), heat treating (as of dies, cutting tools, etc.).

In the Hoskins type F C furnace the current enters by way of one of the electrodes, passes along one side through the carbon plates, through a connector plate, and back to the other electrode through the other carbon plates. The heat is radiated uniformly onto the work. The carbon-plate type furnaces are for work such as for high-speed tool steel where the temperatures required are above 1,900 to 2,000 degrees F.

For temperatures around 1,800 degrees F. the heating element is an alloy wire, chromel. Furnaces of this kind are used for the usual purposes mentioned above and for pre-heating high-speed steel before treating in the carbon plate type or type F C furnace.

By means of pyrometer and regulating rheostat the temperatures are controlled at all times, which makes electric furnaces ideal. A pyrometer with thermo-couple for inserting in the furnace and connecting cord are shown in one of the illustrations.

The regulation of the temperature requires merely a regulation of the current. This is a very simple matter and accounts for the ease and accuracy of control of the electric furnace. Where a rheostat is used, the turning of the rheostat handle provides the regulation. Particularly with larger furnaces, the regulating transformer is preferred, although it can only be used where alternating current is employed. Since low voltage currents are used in the operation of small furnaces, alternating current is preferable, since by means of a transformer the voltage of the usual 110, 220 or 440-volt system can be reduced to the lower voltage required which may be from 10 to 50 volts. Direct-current furnaces are, of course, made and can be used on the usual systems where the voltages are 110 or 220.

The Edison Appliance Company furnace is used as an annealing furnace for treatment of carbon steels, for blueing, annealing and preheating high speed steel and dies, and is used for porcelain enameling of small parts. The operating temperature is about 1,800 degrees F. When put in use about 3,000 watts are consumed, but this is maintained only for part of an hour, when the current consumed drops to less than 1,000 watts which is sufficient to keep the heated furnace up to its proper temperature.

For laboratory and research work the small-tube furnace which operates at a current consumption of 350 watts, and will maintain a temperature of 1,800 degrees F. for a continued period is used extensively. It may be

used for checking pyrometer couples, ash determinations, organic and inorganic combustion work.

Crucible-type electric furnaces are used in determining the decalescent and recalescent point of steel, melting alloys, ash determination and general research laboratory work.—"Industrial Management."

H. M. T. B. DESTROYER *MOUNSEY.*

Destroyers, as is well known, have played a leading part in the defeat of Germany at sea. Not merely did they get home again and again on the German Fleet as it fled from the Jutland battle, but the destroyers have been the terror of the U boats. For obvious reasons it has not been possible to publish until now any details of the more recent of our additions to these craft, but the declaration of the Armistice has relaxed the restrictions in vogue, and we are thus enabled to give an illustration and some particulars of the destroyer *Mounsey,* the boat which, under the command of Lieutenant Craven, saved, under circumstances of very great difficulty, no less than 696 lives when the *Otranto* was torpedoed on October 6 last. The sea at the time was exceedingly rough, and it would have been fatal to have brought a lightly-constructed vessel like the *Mounsey* alongside of the cruiser. The saving of life was therefore effected by maintaining the *Mounsey* under way, so that she passed the *Otranto* within a few feet, allowing the people to jump from one ship to the other. The maneuver had to be repeated many times, and was, under the circumstances, a very original and successful method of rescuing those endangered without great risk to the destroyer herself.

The *Mounsey* was built by Messrs. A. F. Yarrow and Co., Ltd., at their Scotstoun works, and on her measured mile trials attained a speed of over 39 knots. This trial was run with the boat fully armed and equipped, and with sufficient fuel on board for a run of 1,000 miles at an economical speed. Further patriculars are given below.

Length between perpendiculars	260 feet 3 inches.
Length overall	271 feet 6 inches.
Beam	25 feet 7½ inches.
Depth, midships	16 feet 3 inches.
Total heating surface	22,017 square feet.

Four hours' trial—

Draught forward, 8 feet 1½ inches; draught aft, 8 feet 2¾ inches	=835.3 tons at yard.
Speed, 4 hours	38.605 knots.
Speed on measured mile	39.018 knots.
Revolutions per minute, 4-hour trial	685.6
Revolutions per minute on measured-mile trial	693.02.
Oil consumed in 4 hours	57.33 tons.
Load on trial	158 tons.
Oil fuel capacity	228 tons.
Radius of action at full speed	615 miles.

Armament—

Three 4-inch quick fires; two 2-pounders; two twin 21-inch torpedo tubes.
Complement—79.

PARTICULARS OF RUNS ON MEASURED MILES.

No.	Hour. p. m.	Boiler Steam.	Vacuum.	Air Pressure.	Revolutions.	Time on Miles.	Speed.
1	1.45	259	28	6.5	689.0	1-32.4	38.962
2	1.54	257	28	6.5	692.0	1-32.4	38.962
3	2.4	257	28	6.5	693.0	1-32.2	39.056
4	2.13	258	28	6.5	693.5	1-32.0	39.130
5	2.24	258	28	6.5	694.0	1-33.0	38.710
6	2.34	259	28	6.5	692.75	1-31.4	39.388
Mean on measured mile ...	...	258	28	6.5	693.023		39.018
Mean in 4 hours	...	258	28	6.4	685.6		38.605

It may be added the first Italian vessel to enter the port of Trieste was the *Audacé,* which was built by Messrs. Yarrow and Co. eighteen months ago.—" Engineering."

THE COMPOSITION AND PROPERTIES OF STEELS.

By Howard Ensaw.

In most branches of the engineering trade it is at times necessary to reduce the weight of some mechanism, and this applies particularly to the manufacture of automobiles and aeroplanes. This forms probably the chief reason for a thorough study of the nature and properties of the materials employed in the construction of such mechanisms.

It is essential that the designer should have a full knowledge of the strength and durability of all the materials to be used, and it is important that those materials should be of a specified composition, and that the treatment should be checked by suitable tests in order that the results in actual practice shall correspond with the data on which the designer worked.

Until about four years ago the number of firms with a knowledge of the composition of aircraft steels and experience in their heat treatment was most limited, and today there are so many brands of alloy steels requiring different treatment that great care must be exercised. As the hardening and tempering process is liable to error due to the number of brands available, users are well advised to choose only a few types of steel to cover their requirements. The treatment of the selected steels quickly becomes familiar, and the possibility of error is reduced.

A range of about ten steels will cover the requirements of practically every firm, and the object of these notes is to give the composition and properties of a range which might be chosen with advantage.

As variation in heat treatment is required with different steels in order to obtain the best results, it is obviously necessary to consider the mechanical properties along with the chemical composition.

It will be seen that it is possible, however, particularly in the case of nickel-chrome steels, to obtain widely varying results with varied heat treatment.

The table has been compiled from actual tests made with a particular make of nickel-chrome oil-hardening steel, and illustrates this point very well.

The material which gave the above tests when treated in the manner indicated forms one of the most interesting metallurgical studies, and is of exceptional value in aircraft work. It will be noticed that various degrees of hardness can be obtained, and this factor is made great use of in the manufacture of articles where distortion is a disadvantage and at the same time difficult to avoid.

The articles are first rough-machined and then heat-treated to a Brinell hardness of 250 to 321. At this hardness the articles can be machined, and we then have no distortion in the finished goods, as there is no subsequent heat treatment.

O. H. Degrees C.	Temp. Degrees C.	Time. Minutes.	T. S. Tons per square inch.	R. A. per cent.	B. N.
825	650	30	59	57	269
825	600	30	64	48	286
825	550	30	69	42	321
825	500	30	75	43	332
825	450	30	85	42	375
825	450	20	87	49	364
825	400	30	96	38	418
825	350	30	104	33	430
825	200	15	121	38	512
825	Not	tempered	123	34	477
825	Not	tempered	126	21	512
800	Not	tempered	127	27	477

O.H. = Temperature in degrees centigrade at which sample was hardened in oil.

Temp. = Temperature in degrees centigrade at which sample was tempered.
Time = Time in minutes for tempering.
T.S. = Tensile strength in tons per square inch given by test piece.
R.A. = Reduction of area registered during above test.
B.N. = Brinell hardness number.

It will thus be seen that it is possible to obtain widely varying results from one brand of steel, and it might also be said that it is possible to obtain similar results from two steels of widely different chemical composition.

It is highly advisable that the steel manufacturer should coöperate to a greater extent with the steel user to obtain a better understanding of the work for which the finished article is intended. Items subjected to different stresses require different materials and treatment, and in the case of resistance to wear it is well to bear in mind that a steel containing a low proportion of nickel and a low proportion of chromium will resist wear better than a steel containing a high proportion of nickel and no chromium.

It has become generally recognized that it is not only necessary to utilize materials of a sufficiently high yield point, but the materials must also be of a nature suitable for resisting wear if the articles are subjected to abrasion, and able to withstand shock if the articles are subjected to vibration. The advisability of putting samples of the material through tests which bear some resemblance to the stresses which will be met in

actual use is obvious, and in connection with aero-engine manufacture this question has been given considerable attention.

Perhaps the first problem to be attended to in the manufacture of parts which are highly stressed is the question of the composition of the materials employed. It will be understood that if the original material should be of incorrect proportions satisfactory results can scarcely be expected.

It is expensive to introduce the nickel, chromium, manganese and vanadium into mild steel, and unless the alloy steel thus produced is carefully watched its value may be destroyed as the result of incorrect proportions of carbon, sulphur and phosphorus. A full chemical analysis from time to time will probably save the reputation of the firm concerned.

In addition to the expense of manufacture, alloy steels require more careful treatment. A scientifically arranged system of checking temperatures is necessary if the results are to be relied upon. As the employment of these steels is necessary when we require an ultimate strength which may reach 130 tons per square inch, the expense of materials and testing apparatus must necessarily be met.

The elements previously referred to have varying effects, and are introduced one or more at a time. The alloys differ over a wide range, and the treatment of these various alloy steels must be modified to suit each particular composition.

When nickel is added to plain carbon steel, the skin of the hardened material is harder, and the depth to which the quenching effect penetrates is greater.

Chromium and tungsten added to plain carbon steel enable a more highly finished surface to be obtained when machining, and a finer grain is produced in the metal; but these elements do not appear materially to influence the hardness or the depth of case of the hardened material.

Chrome-vanadium steels do not appear to have given quite as successful results as nickel-chrome steels, but for certain purposes they are preferable. The presence of vanadium increases the depth of the hardening effect, but when vanadium is present in even medium quantities the material in the annealed state is very hard, and machining difficulties arise.

Nickel is perhaps the most valuable element to the metallurgist, and by simply normalizing a bar of nickel steel to the same tensile strength as a sample of medium carbon steel a comparison of the results readily establishes the superiority of the alloy steel. Increased elasticity is obtained, and the impact test figures are considerably improved, so that in cases where elasticity and shock-resisting qualities are required, nickel steel is preferable, providing the carbon content is not excessive when the proportion of nickel is considered. If the carbon content is not within the limits specified later, the probability is that the article would fracture under a lighter load than plain carbon steel would stand.

Bearing in mind the influence of these alloys when introduced into steel, we can proceed to inquire into the composition of the materials, and it must be noted that it is necessary to maintain the proportions of the elements within certain limits in relation to each other.

The use of plain carbon steel in aircraft work is most restricted, as we generally require material able to withstand higher stresses; but it can be used for certain parts which are not subjected to great shock or wear, such as tachometer gear. It is then possible to use a steel suitable for automatic machinery, and material to the following specification will be found to give satisfactory results:

Carbon	Not greater than 0.25 per cent.
Manganese	Not greater than 0.85 per cent.

Silicon	Not greater than 0.20 per cent.
Phosphorus	Not greater than 0.06 per cent.
Sulphur	Not greater than 0.06 per cent.

Should it be required to use a slightly higher tensile steel the following proportions will be found satisfactory, but machining is more difficult and the tools will require more frequent attention:

Carbon	Not greater than 0.40 per cent.
Manganese	Not greater than 1.00 per cent.
Silicon	Not greater than 0.20 per cent.
Phosphorus	Not greater than 0.06 per cent.
Sulphur	Not greater than 0.06 per cent.

None of the above steels is suitable for casehardening, but as they can be obtained in bright drawn bars the advantages of their use in all possible instances are obvious.

Although casehardening nickel steel has in a great many cases taken the place of casehardening mild steel, the latter is still used to a great extent in the manufacture of valve tappets, engine-timing gears, camshafts, gudgeon pins, etc. Parts made from this material present a good resistance to wear, and two qualities, low and medium carbon, may be used with advantage. The low carbon will give a minimum of 23 tons breaking strength when normalized at 900 degrees to 920 degrees C., and the composition as follows:

Carbon	0.10 per cent.
Silicon	0.18 per cent.
Manganese	0.60 per cent.
Sulphur	0.04 per cent.
Phosphorus	0.04 per cent.

When this material is normalized as above the physical tests should be as follows:

Breaking strength	23 to 28 tons per square inch.
Yield ratio	Not less than 50 per cent.
Elongation	Not less than 30 per cent.
Reduction of area	Not less than 50 per cent.
Brinell hardness	92 to 112.

This material if well made will be found to caseharden well and uniformly, but it is frequently necessary to employ a similar steel with slightly improved breaking strength, although there is a tendency for the elongation to diminish.

This second-quality casehardening mild steel can be specified as follows:

Carbon	0.15 per cent.
Silicon	0.18 per cent.
Manganese	0.75 per cent.
Sulphur	0.06 per cent.
Phosphorus	0.06 per cent.

It will be noticed that higher proportions of sulphur and manganese are permissible in this instance. When the above material is normalized at 890 degrees to 920 degrees C., the following tests should be obtained:

Tensile breaking strength	25 to 33 tons per square inch.
Yield ratio	Not less than 50 per cent.
Elongation	Not less than 25 per cent.
Reduction in area	Not less than 50 per cent.
Brinell hardness	103 to 143.

The normalizing temperatures given are within 50 degrees C. above the critical temperature in each case. The carburizing should take place at 900 degrees C., the length of time depending upon the depth of case required, and the articles should then be allowed to cool down in the compound. In order to refine the structure of the core the articles should then be heated to about 870 degrees C. and either allowed to cool in the air or quenched in water. Finally they should be reheated to 780 degrees C. and quenched in water to harden the case.

With certain makes of steel not conforming correctly to the above specifications it may be necessary to depart very slightly from the above temperatures, but the most suitable conditions can readily be ascertained by trial and a little advice from the steelmaker.

There are many instances where it is necessary to have some knowledge of the mechanical properties of the core of casehardened articles, and this can be obtained by allowing test pieces to go through the same heat treatment and at the same time as the parts being manufactured, taking care to turn off the case to the depth to which the carbon has penetrated. It will generally be found that by the refining influence of the heat treatment the yield point and the breaking strength are improved, and the elongation and reduction of area are reduced.

In carrying out some recent tests in connection with the last material detailed, the tests on the material as rolled and after the hardening treatment were as follows:

	As Rolled.	After Treatment.
Breaking strength	31 tons	36 tons.
Yield point	21 tons	25 tons.
Elongation	28 per cent	23 per cent
Reduction of area	58 per cent	51 per cent.

These figures are of some value, as it must be realized that we cannot count on the case to assist in withstanding any shocks, and our calculations must take into consideration only the extent to which the material may be loaded based on the strength of the core only.

Should circumstances arise where it is necessary to employ material of higher ultimate strength and better surface-wearing qualities than the casehardening carbon steels detailed above, use can be made of casehardening nickel steels, and here it is necessary to exercise greater care in manufacture and closer accuracy in heat treatment.

A good low-nickel alloy steel can be made up as follows:

Carbon	0.13 per cent.
Silicon	0.25 per cent.
Manganese	0.45 per cent.
Sulphur	0.04 per cent.
Phosphorus	0.04 per cent.
Nickel	2.00 per cent.

The carbon content must be carefully checked, as, if excessive, this causes shortness, and attempts to straighten any parts distorted in hardening will result in fracture.

This material when normalized at 850 degrees to 900 degrees C. should give the following results:

Tensile breaking strength	25 to 35 tons per square inch.
Yield ratio	Not less than 55 per cent.
Elongation	Not less than 30 per cent.
Reduction of area	Not less than 55 per cent.
Brinell hardness	103 to 153.

A step higher in tensile breaking strength and yield ratio can be obtained with an alloy steel containing a larger proportion of nickel, and the elongation will remain about the same. A chemical analysis of such a steel should show the results given below, and the necessity for maintaining a low proportion of the injurious elements renders the manufacture of this steel somewhat difficult and consequently expensive:

Carbon	0.15 per cent.
Silicon	0.18 per cent.
Sulphur	0.04 per cent.
Phosphorus	0.04 per cent.
Manganese	0.35 per cent.
Nickel	5.00 per cent.

The casehardening nickel steels given above are suitable for articles where a good hard wearing surface is necessary, but when quenching to harden the case it is necessary to reheat to a somewhat lower temperature than is the case with carbon steels, and when it is decided which brand of material to use it is advisable to carry out a series of tests to ascertain the most suitable temperature. Occasionally it is found necessary to manufacture articles where good wearing qualities are required, but the parts are so thin that a hard case would be disastrous, as there would be no core left. In such instances an alloy steel containing about 3 per cent of nickel, 0.30 per cent of carbon, and about 0.70 per cent of manganese will be found of great value, as it is thus possible to obtain 45 tons per square inch ultimate stress and a Brinell hardness of about 200.

The materials which have probably been of greatest value to the automobile manufacturer engaged on aero-engine work are probably the alloy steels containing nickel and chromium. These are very high in tensile strength, and are suitable for oil-quenching and tempering or air-hardening according to the proportions of nickel and chromium in their composition.

These alloys can be so constructed that they can first be hardened and then tempered to give almost any degree of hardness between 250 and 500 Brinell, and it is not a difficult matter to bring the hardness to about 320 Brinell. Up to this figure the material can be machined, and the parts will stand a great deal of shock and wear; but should the hardness be greater, the probability is that the further machining would be impossible.

It must not be concluded from the above remarks that the Brinell hardness mineral can be taken as a general guide to the machining properties of materials, but in comparing two samples of the same steel it can be taken as a rough indication. When comparing different alloys it is quite possible that one steel can easily be cut by another which registers lower on Brinell test.

A high tensile nickel-chrome steel suitable for tempering to various degrees of hardness can be built up as follows:

Carbon	0.20 to 0.30 per cent.
Silicon	Not over 0.30 per cent.
Manganese	0.35 to 0.60 per cent.
Sulphur	Not over 0.04 per cent.
Phosphorus	Not over 0.04 per cent.
Nickel	2.75 to 3.50 per cent.
Chromium	0.45 to 0.75 per cent.

If the above material be heated up to 820 degrees C. and quenched in whale oil, then tempered at 600 degrees C., the test results should be approximately:

Tensile breaking strength	Not less than 45 tons per square inch.
Yield ratio	Not less than 5 per cent.
Elongation	Not less than 15 per cent.
Reduction of area	Not less than 50 per cent.
Brinell hardness number	179.

When using this material for a high tensile and a hardness of about 320 Brinell as previously suggested, it is an advantage to rough-out the parts before heat treating, so that very little metal is left for removing in the hard state.—"Mechanical World."

NEW TYPE OF MARINE OIL ENGINE.

Mr. Carl W. Weiss, associated with the Metz & Weiss Engine Company for a number of years and now connected with the Weiss Engine Company, 17 Battery Place, New York, has developed a surface-ignition, medium-compression, two-cycle oil engine that promises to widen the scope of the heavy oil motor. The engine illustrated herewith is a 400-brake horsepower, 4-cylinder unit, although the type is also being built with six and eight cylinders. Among other features it possesses a new method of scavenging, and, in this connection, when it is recalled that the ordinary type of surface-ignition, medium-compression oil engine uses a baffle plate in conjunction with crank-case compression for scavenging—a construction which, due to uneven thicknesses of metal, makes for uneven contraction and expansion that frequently results in cracked piston heads—it is well to note that in the accompanying illustration the piston is conical in shape and a rather wide departure from usual practice.

BONE OF CONTENTION.

The bone of contention regarding the relative efficiency of the surface-ignition, medium-compression, two-cycle oil engine has always been the matter of scavenging. That previous existing types of this style engine have not completely burned their charge or completely cleared the cylinder of burned gases after each explosion has been the contention. In the engine here shown there has been incorporated an entirely new method of scavenging. There are three annular sets of piston-controlled ports: (1) the exhaust, (2) the supplementary, and (3) the crank-case port. The supplementary ports are open to either atmospheric or under low pressure of air supplied by a small pressure blower. As the piston uncovers the first series—the exhaust ports—near the end of the expansion stroke, the pressure of the cylinder drops to atmosphere, and, due to the abrupt discharge and the forcible cooling of the gases, the pressure at once goes

down to several points below atmosphere. At this point the supplementary ports open, allowing a charge of pure air to sweep in radially over the conical piston head, displacing the exhaust gases left in the cylinder, while immediately following this, as the crank moves through the lower dead center, the crank-case air under approximately five pounds pressure per square inch also flows in over the conical piston head by way of the annular series of ports formed by the spirally ribbed lower parts of the cylinder liner. In this way three completely separate and distinct charges of air are introduced into the cylinder during the scavenging process, which, undoubtedly, accounts for the fuel efficiency of this new type and its ability to operate indefinitely without undue heating of piston head.

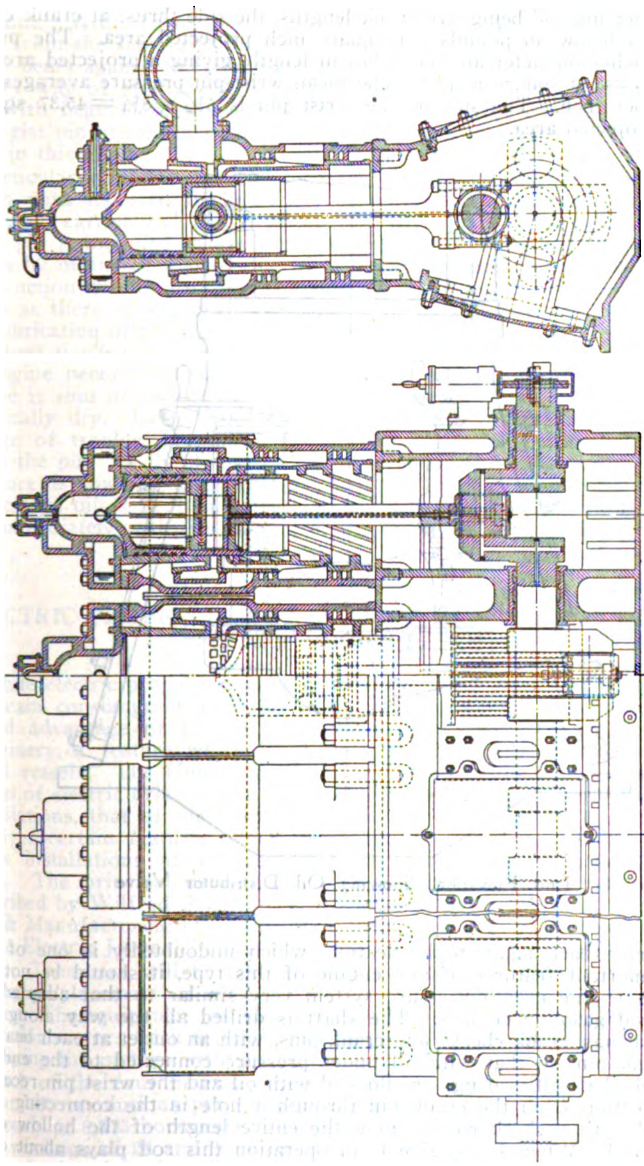
SIMPLE OIL INJECTION SYSTEM.

The oil-injection system of this multi-cylinder engine is simple. In place of direct-driven governor-control injection pumps, there is an independent duplex pump to keep the oil under constant high pressure, and a compensating distributor valve, driven from the engine shaft, arranged for timing adjustment for different grades of oil and either direction of rotation. This pump is connected to the air receiver used for starting and reversing the engine. With a normal air pressure of 200 pounds in the receiver, the oil pressure is kept at 1,000 pounds by a reducing valve in the air line. Heavy oils require high pressures for efficient spraying. The governor is designed to act directly on the compensating valve and is, in fact, carried by the distributor valve and submerged.

There is a spiral gear mounted on the front end of the crank shaft which drives the oil distributor on one side and the air distributor for starting and reversing on the other. Each cylinder has an air check valve piped to the air distributor and a relief valve open to the atmosphere. These relief valves can be operated either independently or simultaneously by a lever at the front end of the engine, so that the entire control of speed, starting and reversing, and pressure relief is brought within easy access of the engineer.

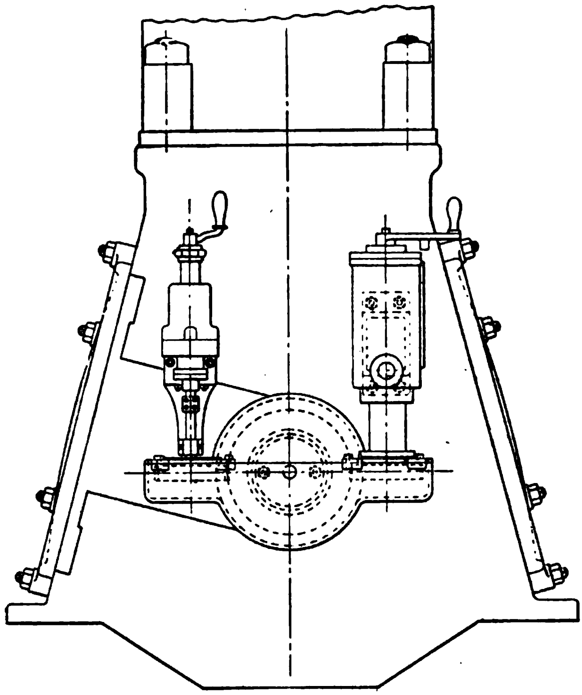
Forced-feed lubrication is used for the cylinder, main bearings, crank pin and wrist pin, each pipe terminal fitted with a lubricating sight check. These lubricators are of the single-plunger, distributor-disc type, furnished by L. T. Weiss, of Brooklyn, and used in large quantities by the United States Government on single and multi-cylinder engines and guaranteed to force oil against 200 pounds pressure. A hardened steel worm engages a worm-wheel disc, which latter carries a steel plunger with its operating yoke. As the check turns, the two diametrically opposite projections of the yoke and the fixed star wheel operate as an escapement, reciprocating the plunger, to draw in and discharge oil through a hole in the disc, registering alternately with the suction and discharge hole in the base. Each discharge has a screw coupling for copper tubing. The whole mechanism being very simple and substantial, and submerged in oil, with an extremely slow movement, these lubricators run for many years without the least wear or adjustment.

This type of marine engine requires no flywheel and no special scavenging pumps. The weight per horsepower is reduced to approximately 150 pounds without sacrificing a reliable factor of safety within the limits of working pressure. At a pressure of 500 pounds per square inch the maximum main bearing pressure does not exceed 509 pounds per square inch projected area. The shaft in the 16½ by 22 unit is 8¼ inches diameter. The bearing length is 12¾ inches. There is a center bearing between each cylinder in the multi-cylinder type. The bearing pressure, therefore, is $500 \times 214 = 107{,}000$, divided by $210 = 509$ pounds, as stated above, while



Elevation and Cross Section of Weiss Engine, Showing Conical Piston and Cylinder Liner, Together With Method of Supporting Main Bearings from Side

the mean pressure is below 100 pounds per square inch projected area. The mean crank-pin pressure equals 290 pounds per inch projected area. The connecting rod being five crank lengths, the side thrust at crank circle tangent is below 40 pounds per square inch projected area. The piston is 16½ inches diameter and 33 inches in length, giving a projected area of $33 \times 16\frac{1}{2} = 544$ square inches. The mean wrist-pin pressure averages 475 pounds per inch. The size of the wrist pin is $5\frac{1}{2} \times 8\frac{1}{4} = 45.37$ square inches projected area.



End Elevation, Showing Oil Distributor Valve

Referring back again to lubrication, which undoubtedly is one of the most important features of any engine of this type, it should be noticed that this engine uses a pressure system very similar to that adopted by high-speed gasoline engines. The shaft is drilled all the way along the main bearings, crank cheeks and crank pins, with an outlet at each bearing and crank pin, so that with oil under pressure connected to the end of the crank shaft all bearings are flooded with oil and the wrist pin receives its lubrication from the crank pin through a hole in the connecting rod, provided with a check rod running the entire length of the hollow connecting rod. When the engine is in operation this rod plays about one-eight of an inch between the wrist pin and the crank box for the purpose of checking the oil which has once passed the rod, and retaining same for wrist-pin lubrication. With this pressure system there is really no need for a special check rod for the oil in the connecting rod, but this provision

is of considerable advantage, inasmuch as there is no oil in the rod when the engine has been standing for a sufficient time to enable the oil to leak out. With the oil in the rod the wrist pin will get its lubrication right from the start, and this prevents any cutting of the bronze bushing which bears against the steel-hardened and ground wrist pin. In the customary way of putting the wrist pin directly through the piston, usually cast with heavy bosses on each side, the chances of conducting heat to the wrist pin are much greater than in the wrist-pin carrier arrangement used in this engine.

Particular attention is drawn to the illustration showing the conical piston with its even thickness of metal, to the connecting rod, to the piston-pin carrier and to the piston pin, where it will be noted that the wrist pin is mounted in a separate carrier which is locked inside of the piston by means of a snap ring, thus dispensing with the ordinary piston construction calling for heavy bosses on the body of the piston. Inasmuch as there is less heat conducted with the arrangement here shown, the lubrication of the wrist pin and durability of it is materially increased. The heat flowing from the piston wall to the wrist pin in the other style of engine necessarily makes the lubricating oil very thin, and when the engine is shut down and the lubricating-oil feeds ceased, the pin becomes practically dry. Later, when the engine is started, there is a very good chance of trouble, because it takes several minutes before the oil can reach the pin. In the design of wrist-pin carrier here set forth, the temperature is lowered and, with a check rod in the connecting rod, the lubricating oil is kept at a level which will provide lubrication for the pin immediately with the starting of the engine.—"Marine Engineering."

ELECTRIC PROPELLING MACHINERY FOR THE BATTLESHIP *TENNESSEE.*

While electric propulsion gives approximately the same overall economy in steam consumption as a gear-turbine drive, nevertheless it offers an added advantage of greater latitude in the location of the propelling machinery, a feature which is of greatest importance in the design of naval vessels. The United States Navy has taken the lead in the application of electric drive to battleships and battle cruisers; and of the recent installations, that of the battleship *Tennessee* is of special interest, as it involves certain features which have been developed as a result of previous installations of electric propelling machinery in vessels of this class. The principal details of the installation on the *Tennessee* were described by Wilfred Sykes, general engineer for the Westinghouse Electric & Manufacturing Company, East Pittsburgh, Pa., in a recent issue of "The Electric Journal," from which the following information is taken:

The *Tennessee* will be one of the most powerful fighting ships built, having displacement of over 32,000 tons and a speed of 21 knots at full power. The equipment for propelling the ship will consist of four 3-phase induction motors, each driving one propeller, and two turbo generators for supplying the motors with power.

Each of the four motors will develop 7,000 horsepower at a speed of about 180 revolutions per minute, and will be capable of working continuously at 8,375 horsepower as an overload condition. The motors have two windings of 24 and 36 poles, so that they have two normal speeds of 123 and 180 revolutions per minute with full speed of the turbine. In this way it is possible to run the turbine at its most economical speed when steaming either at full power or cruising at 15 knots. Intermediate speeds are obtained by varying the speed of the turbine, and the equipment is

designed to maintain a low water rate over the full range of speed from ten knots up. When operating below 17 knots, only one generator is used, and this improves the economy, as the load on the unit is brought nearer to its full capacity.

TWO-POLE GENERATORS.

The turbo generators supplying power to the main units each develop 13,500 k. v. a. at full speed and are capable of carrying 15,000 k. v. a. continuously for the overload conditions. The generators are two-pole machines and the unit runs at 2,190 revolutions, corresponding to 36.5 cycles per second, with the motors running at 180 revolutions per minute. The maximum speed of the turbo generator is 2,270 revolutions per minute, corresponding to 37.9 cycles, equivalent to a motor speed of 186.5 revolutions per minute, which requires 8,375 horsepower. To obtain lower speeds, the turbine speed is reduced to about 1,500 revolutions per minute, which corresponds to the change-over point from the 24 to 36-pole connection of the motors.

With the change-over of the motor connections, the speed of the generator is increased to 2,270 revolutions, corresponding to 15 knots with the 36-pole connection. The motor speed combination is simply the equivalent of a variable ratio of gearing, which, in the case of the 24-pole connection, is 12 : 1, and with the 36-pole connection 18 : 1.

The direction of rotation of the machines is controlled by reversing switches which simply transpose two of the phases of the motors, the generator, of course, continuing to run in the same direction.

The motors have two separate windings on the stator, one the 24-pole and the other the 36-pole connection. The same results might have been obtained by use of one winding, but this would have entailed greater complication in the connections and would have restricted the design in other ways.

The rotor has a single polar winding for 24 poles, which is connected to the slip rings in the ordinary way, so that resistance can be inserted in the circuit during starting or reversing. When the machine is operating on the 36-pole connection the rotor winding cross-connections act as short-circuiting connections for this pole combination. With the 24-pole combination, they act as equalizing connections between points of equal potential. On the 36-pole connection the motor operates as a squirrel-cage machine, and it is not intended that this winding should be used during the starting or reversing, but only as a running winding. In this way only one winding is used and one set of slip rings.

THE TURBINE GOVERNOR.

The speed of the turbine is varied by means of a unique hydraulically-operated governor. The loading of the governor is regulated by means of a variable-pressure oil system, the pressure of which is regulated with great accuracy by a pressure regulating mechanism operated by the control handle. In this way any mechanical connection through shafts or rods with the governor from the operating point, with the consequent danger of jamming where passing bulkheads, is avoided. A unique feature of this arrangement is that the pressure is caused to pulsate slightly so that the whole of the regulating and governor mechanism is kept slightly in motion, and thereby prevented from sticking, also adding greatly to the sensitiveness of the control, which is of great importance when ships are steaming in formation.

The turbines are of the Westinghouse semi-double flow, impulse-reaction type. The high pressure steam is expanded in suitable nozzles and

passes through a two-row impulse wheel, after which it passes through the first stage of the reaction expansion, which is single flow. The steam then divides and passes through the low-pressure stages of the turbine, which are double flow. The turbine is provided with an automatic stop to cut off steam in case the speed should exceed the maximum safe operating value. The main hydraulically operated governor maintains speed practically constant at any value set by the control mechanism, independent of the load, so that in case the propellers should leave the water during rough weather there will be no racing.

INSULATION.

The generators and motors are very carefully insulated for this service, so as to prevent damage due to moisture or the accumulation of salt, and also due to the high temperatures which are liable to be encountered in this service. The principal material used for insulation of coils in the slots is mica, and the machines are capable of withstanding slot temperatures up to at least 150 degrees C. without injury. Ventilation of the generators is provided by fans supplying air to each engine room and the fans on the generator forcing the air through the machine and out through ducts. The motors each have two fans mounted directly above them which draw the air through the motor and force it out through the ventilating ducts. The generators are excited from the direct-current power circuit of the ship through boosters which are capable of raising the normal 240-volt supply to 320 volts or reducing it to zero.

The power supply from the turbo generators is brought to a centrally-located control room in which is mounted all the necessary switching apparatus for controlling and distributing the power to the motors. In this room is mounted the regulating apparatus for the main turbines, the field switch and rheostat for the turbo generator excitation, and the liquid rheostats for the main motors. All necessary instruments for the operation of the equipment are mounted directly in front of the operators and full advantage is taken of the great facility with which electric power can be measured, inasmuch as it will assist in the operation of the ship.

AUTOMATIC LIQUID RHEOSTATS.

For starting and reversing the main motors, automatic liquid rheostats are used which are of a similar design to those used previously for industrial purposes. These liquid rheostats consist of two tanks, the upper containing a series of fixed electrodes and the lower acting as a reservoir. By means of a suitable pump, the electrolyte is caused to flow from the lower to the upper tank at the proper rate to cause the desired acceleration. When the by-pass between the two tanks is open the electrolyte is maintained within the electrode tank at the proper level to give the maximum resistance. When this by-pass is closed the electrolyte rises in the upper tank, thereby progressively short-circuiting the electrodes until the minimum resistance is reached at the overflow point, after which the liquid simply continues to circulate through the two tanks. A switch is provided so that this rheostat may be short-circuited.

The cables for connecting the turbo generators and motors are of great importance, as the operation of the ship depends upon their reliability. A number of parallel circuits are used, each cable being of the three-core type, and the failure of any single cable would not seriously interfere with the operation of the ship. The main cables are of the same order of importance as the main steam pipes; hence the greatest care has been taken to insure that these cables should be the best type that is possible

to manufacture for the service, and to this end a committee of the American Institute of Electrical Engineers assisted the Navy Department in preparing the specifications.

ENGINE-ROOM AUXILIARIES.

The main auxiliaries in the engine room are electrically driven. The main circulating pumps are driven by 235-horsepower, direct-current motors, directly connected to centrifugal pumps, the speed of which can be varied to suit various conditions of operation. In this way the power consumption can be reduced as the speed of the ship is reduced. The principal provision for maintaining vacuum in the condenser is the use of LeBlanc air ejectors, which are novel for this class of ship. These air ejectors have already been successfully tried out by the Navy Department on other vessels with such satisfactory results as to justify their adoption for these vessels. The condensate from the condensers is handled by vertical electrically-driven centrifugal condensate pumps, so that the whole of the essential auxiliary apparatus for the turbines is rotary, and, based on the experience in land service as well as at sea, a greater reliability and lower maintenance can be anticipated for these equipments compared with past practice. The use of air ejectors enables the vacuum to be maintained at least as high, if not higher, than the older combination of reciprocating air pump and Parsons augmentor, and the space and weight are a very small fraction of that required with the older system.

While steam consumption is not of vital importance, on account of the other advantages of electric drive, yet the figures that can be obtained are very appreciably better than the direct-connected turbine and are lower than any past practice. From 10 to 15 knots only one generator is used, the motors being connected for 36 poles. From 15 to 17 knots the motors are connected for 24 poles and one generator is used. From 17 knots to 21 knots both generators are in operation, each machine supplying power to two motors, each side of the ship being independently operated.

In designing the equipment for the *Tennessee,* every effort has been made to avoid the introduction of experimental or risky constructions, and the design is such that the experience gained in other fields has been utilized to full advantage, and the design factors have been kept within well-developed practice. At the same time, provisions have been made so that the full advantages of the characteristics of electric drive can be utilized in the operation of the ship.—"Marine Engineering."

SEVEN WORKSHOP HINTS.

By Fred Horner.

Repairing Broken Hack-saws.—Fig. 1 shows a method the writer has found very effective in dealing with hard hack-saw blades which have broken, and need a fresh attachment in a shortened frame. It is difficult and time wasting to attempt softening and drilling for a fresh hole, and a far quicker dodge is to run a groove into the saw with a narrow grinding wheel. This takes but a few seconds, and the heat developed softens the blade in the area, so that it is better able to resist breakage than if hard. The slot is then hooked over the pin and the frame strained up as usual.

Soft Jaws for Pliers.—When soft brass or other material has to be bent with the pliers, the hardened serrated jaws of the latter damage the surface of the metal. Fig. 2 shows how to fit readily detachable clams,

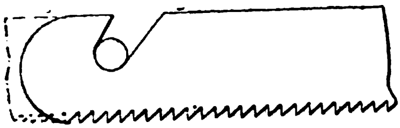


FIG. 1.

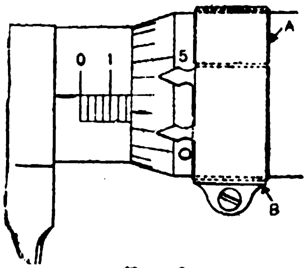


FIG. 3.

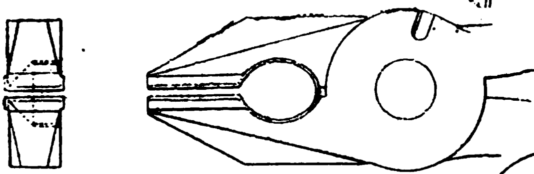


FIG. 2.

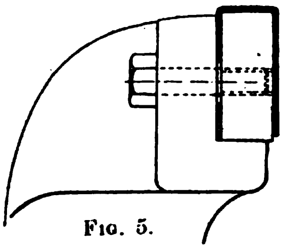


FIG. 5.

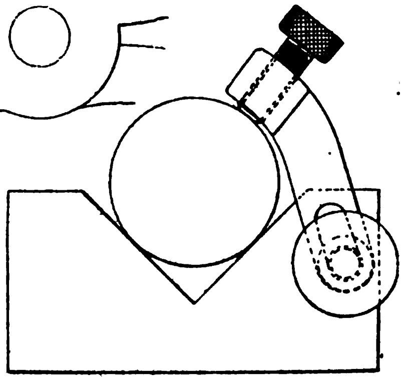


FIG. 4.

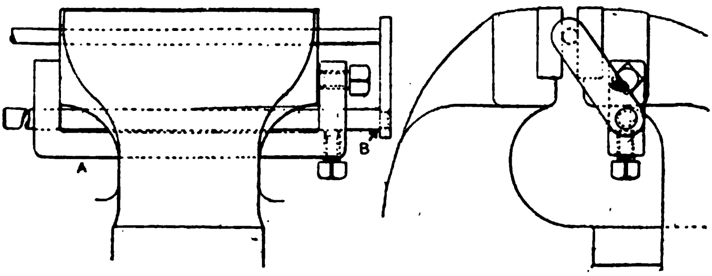


FIG. 6.

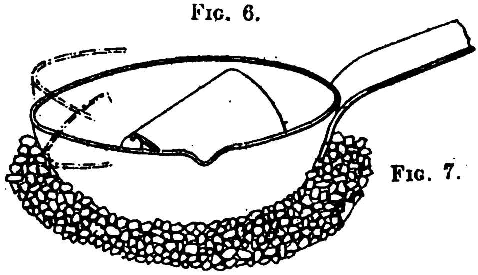


FIG. 7.

made of a strip of soft copper, looped to catch in the concavity between the jaws, and flanged to fit over the nose and prevent side movement.

Limit Fitting for Micrometer.—Fig. 3 represents a fitting to lessen the trouble of reading a micrometer caliper on repeat sizes, incorporating a pair of arrows or pointers which can be set opposite to the high and low limits respectively, thus drawing attention to the graduation lines in a clear fashion. One band of thin metal A, with a pointer, is encircled by another B, furnished with ears and screw and nut to draw it tightly around the inner one. This screw is tightened after twisting the bands respectively to bring the pointers to the graduations desired.

Clamp for Marking-off Vee-block.—Fig. 4 shows a clamp attached to the side of a vee-block, and used to bind a shaft with sufficient firmness for marking-off purposes. It is more particularly useful in cases where a shaft or a casting or forging has arms or lumps upon it.

Attaching Clam to Vice Jaw.—A neat method of attaching a brass or copper clam to a vice for some work which lasted several days was noticed by the writer, and is illustrated in Fig. 5. The holding screws of the steel jaws are slackened sufficiently to admit the interposition of the bent round corner of the clam, and again tightened, giving a grip that permits of no slipping about.

Stop for Vice for Cutting Off Rods.—A fitting for a bench vice to hold a stop-rod, without any necessity for drilling or otherwise mutilating the vice, may be seen in Fig. 6. The device comprises a clamp A straddled over the back jaw, and locked with a set-screw, and holding a rod B. The latter has a plate riveted on it, and the rod is adjusted so that when the work is projected to the correct distance, a hack-saw may be applied close to the side of the vice.

Adjusting Work for Soldering or Brazing.—The final illustration shows a very useful tip which has been found of much value in supporting articles of awkward shape for soldering or brazing. The work (in the example, a blade for an air blower) is rested within a ladle, and the latter is supported upon a bed of small coke. By tilting the ladle suitably any portion of the work can be brought level to let the solder or spelter flow exactly where wanted. The ladle also conserves the heat, and saves the metal dropped from the joint.—"Mechanical World."

MEMORANDUM ON CUTTING LUBRICANTS AND COOLING LIQUIDS, AND ON SKIN DISEASES PRODUCED BY LUBRICANTS.*

PART I.—CUTTING LUBRICANTS AND COOLING LIQUIDS.

1. Cutting lubricants and cooling liquids are oils or emulsions used in connection with the cutting of metal. They possess lubricating and cooling properties in different degrees, and the various classes into which they are divided may be defined as follows:

Soluble Oils.—The products known as soluble oils are only liquids which form an emulsion when mixed with water.

Soluble Compounds, Also Known as Cutting Compounds.—Soluble compounds or cutting compounds are greasy pastes which form an emulsion when mixed with water.

Cutting Emulsions.—Cutting emulsions are aqueous emulsions formed by mixing soluble oils or soluble compounds with water.

*Bulletin No. 2, of the Scientific and Industrial Research Department. Reprinted by permission of the Controller, H. M. Stationery Office.

Cutting Oils.—Cutting oils are oils such as lard oil, rape oil, or mineral oils, or a mixture of such oils free from water and from soap. These oils do not ordinarily form emulsions with water.

2. Cutting lubricants and cooling liquids are used for the purpose of—

(*a*) Cooling.
(*b*) Lubrication.
(*c*) To produce smooth finish.
(*d*) To wash away chips.
(*e*) To protect the finished product from rust or corrosion.

(*a*) *Cooling.*—During operation the heat developed warms not only the tool, but also the material which is being machined. On cooling the latter will contract, and the dimensions will differ from the measurements taken during the process of machining.

The importance of properly cooling the product is, therefore, obvious, particularly under high-speed conditions and with materials, such as aluminum, which have a high coefficient of expansion.

If the tool heats too much the cutting edge will wear rapidly. The heat generated at the point of the tool is conducted into the body of the tool; if the tool is of large section the heat is more readily dissipated than is the case with a tool of light section. Efficient cooling of the tool edge reduces wear and enables a greater output to be obtained. This is most apparent with high-speed steel, the gain in cutting speed on steel and wrought iron being from 30 to 40 per cent, and on cast iron from 16 to 20 per cent. Efficient cooling of the shavings on the side not in contact with the tool is particularly important with tough material, helping to reduce the friction produced by the shavings rubbing against the nose of the tool.

(*b*) *Lubrication.*—Lubrication is of little importance where the manufactured article is made of brittle material, as the material is removed in the form of powder or fine chips. In the case of cast iron considerable advantage may be obtained by using an aqueous emulsion in order to wash the dust away from the working parts and to prevent its dispersal in the air.

Lubrication is very important where the metal is tough, the material being removed in the form of spiral shavings, which grind their way over the nose of the tool. The character of the chips or shavings produced will depend upon the form given to the tool by grinding, and also upon the angle at which it is used.

The heavier the cut the greater will be the metallic friction and the greater the necessity for lubricating the nose of the tool, otherwise the shavings will produce great friction, resulting in rapid destruction of the tool and in rough finish.

(*c*) *To Produce Smooth Finish.*—When the requirements of cooling and lubrication are satisfied the product will receive a good finish. It is also desirable that the cutting oil should have sufficient fluidity to ensure a rapid stream being concentrated when required. Where a perfect finish is desired, experience has shown that cutting oils possessing great oiliness must be applied. For this reason various animal or vegetable oils, or rich mixtures of such oils with mineral oils, are usually employed. Some engineers find vegetable oils possessing great oiliness, such as rape or cotton seed oil, preferable to either mineral or animal oils in producing a very smooth finish. Dies, taps, reamers, and form-tools have a longer life when used on tough steel if a cutting oil is employed in place of an emulsion prepared from a compound or soluble oil. For finish boring, rifling, etc., a mixture of castor oil and mineral cleaning oil (gravity about 860—890) in the proportion of 3 parts of cleaning oil to 1 of castor oil has been used with good results; although those oils do not form a

homogeneous mixture, the addition of an equal volume of turpentine substitute (white spirit) causes perfect solution to take place, and is said to be advantageous for finish-turning on guns and hard material.

(*d*) *To Wash Away Chips.*—Frequently the washing away of chips is quite an important function of the cutting lubricant or cooling liquid, particularly in cases of deep drilling, as in drilling rifle barrels and the like, also in most milling operations.

If the cutting emulsion is used too weak it will not carry away with it the minute particles of metal and scale which may prove detrimental to the machine tool.

In the boring of deep holes, gun tubes, etc., a solution of sodium carbonate (50 pounds) and soft soap (25 pounds) in water (200 gallons) has been found to give very satisfactory results.

In solid deep-hole boring, where cutting emulsions are used, it is sometimes found that the emulsion, in filtering through the chips in the bore, becomes changed in character in such a manner as to lose some of its lubricating quality.

(*e*) *To Protect Finished Product from Rust and Corrosion.*—Good cutting oils used "straight" (*i.e.,* not emulsified with water) will not cause rusting.

Cutting oils containing fixed oils (animal or vegetable oils) such as lard oil, with a large percentage of free fatty acid, will cause verdigris on brass parts. Vegetable oils containing only a small percentage of free fatty acid, such as rape oil, when employed in cutting oils do not produce verdigris unless the oils are rancid.

Cutting emulsions made up from cutting compounds or soluble oils and water cause rusting if they are used too weak, or if they contain acid. Water from some sources contains considerable quantities of sodium chloride (common salt), which is most destructive to the emulsion; in such cases a supply of water free from salt must be used.

Emulsions of oil and water are not stable in the presence of even minute quantities of acid. The acid causes separation of the emulsion into layers of oil and water, the water is circulated by the pump and causes rusting of the work. To a limited extent the emulsion can be reformed by adding a calculated quantity of ammonia sufficient to neutralize the acid, but any excess of alkali may facilitate corrosion of the metal being worked.

The admixture with a soluble oil of paraffin oil (5 per cent or more) prior to the addition of water has been reported to give good results; a thin film of paraffin oil forms on the top of all standing oil in barrels and tanks and prevents the access of air; similarly a thin film of paraffin oil forms over machined parts, machines and tools, which prevents gumming and rust. It should be noted, however, that the addition of paraffin to a soluble oil reduces its lubricating and emulsifying properties.

Emulsions must not be made by mixing soluble oils or cutting compounds with hard water owing to the precipitate caused by the action of the calcium and magnesium salts in such water. Soft water must be used, which may be either rain water or distilled water, or, if neither of these is available, the hard water must be softened by chemical means.

SELECTION OF CUTTING LUBRICANTS.

3. Before selecting the correct grade of cutting lubricants it is necessary to consider several important factors such as—

(*a*) Cutting speed and depth of cut.
(*b*) The material under manufacture.
(*c*) The system of application of the lubricant or emulsion.
(*d*) The production of skin diseases.

(*a*) *Cutting Speed and Depth of Cut.*

Low Speed and Shallow Cut.—Low speed and shallow cut requires little cooling and little lubrication.

Low Speed and Heavy Cut.—Low speed and heavy cut, particularly if the material is tough, demands a cutting lubricant possessing great oiliness.

High Speed and Shallow Cut.—High speed and shallow cut demands a cutting medium with great cooling properties, consequently emulsions are frequently used. Where a perfect finish is desired, low viscosity cutting oils are used "straight."

Where the speeds are particularly high emulsions only should be used, as otherwise there will be excessive heating of the tools and of the product.

Turpentine substitute ("white spirit") is a satisfactory lubricant for aluminum, but it possesses the undesirable quality of inflammability, and if employed must be used with care on this account.

A mixture of petroleum burning oil (paraffin oil) with lard oil or other cutting oil for high-speed work in connection with aluminum is also dangerous and has led to several fires. It is better to use cutting emulsions which possess the necessary cooling properties and are not inflammable.

High Speed and Heavy Cut.—High speed and heavy cut demands a cutting lubricant with great cooling as well as lubricating properties, so that heavily compounded cutting lubricants of low viscosity must be used. Low viscosity is necessary to give good cooling effect, and heavy compounding with animal or vegetable oils is requisite so as to lubricate the tools and shavings effectively and prevent wear as far as possible. For tough material and heavy cuts an emulsion containing 15 to 20 per cent of a vegetable oil has been reported to be satisfactory.

(*b*) *Material Under Manufacture.*—The influence of the material upon the choice of cutting oil has already been referred to (see Section 2).

Where material is brittle, cutting emulsions are nearly always used.

Where the material is tough, but where the speeds are high and the cut light, cutting emulsions are also frequently employed; but where the material is tough and the cut heavy it is necessary to employ cutting lubricants used "straight" and containing a percentage of animal or vegetable oils (ranging from 10 to 50 per cent) or consisting entirely of such oils.

With automatic screw-cutting machines, emulsions have been found in some cases to form a deposit in the working parts of the machines. This may be avoided by the employment of "straight" oils, but in view of the present scarcity of oil, it is advisable in the national interest to use emulsions wherever possible.

The amount of soluble oil or soluble compound used for preparing the cutting emulsion varies from 2½ to 20 per cent, the richer mixtures being used for severe conditions, and the weaker mixtures for light duties or for materials such as brass and aluminum where there is no danger of rusting.

(*c*) *System of Application.*

The cutting lubricant may be applied by hand by a drop-feed system or by some system employing gravity or pumping.

In large machine shops the cutting oil is sometimes circulated through pipes throughout the works, returning through other pipes from the machine tools to a central tank.

Group systems with central tanks are excellent where one mixture is used for all machines on the circuit.

Return pipes should be large, and should be arranged for easy access for cleaning; in large systems isolating valves should be employed to sectionize the system; efficient strainers should be fitted on all return pipes and pump sections and should be cleaned daily. Tanks as a rule should

be cleaned out every four weeks and return pipes every four months. Any scum formed should be skimmed off the tanks daily.

It is important both with this system and where machine tools have individual pumps that the pump's suction should be always covered so that air cannot be drawn into circulation, since aeration of the circulating medium has a strong oxidizing effect upon the oil or emulsion. The important point to keep in mind in regard to the system of application of the lubricant is that where the oil or emulsion is circulated over and over again, it is exposed to the oxidizing effect of air and to admixture with dust and dirt from the machine shop.

A certain amount of oil or emulsion is always splashed away from the machine tools in the form of spray, and is lost notwithstanding precautions taken by the use of splash guards; the loss of oil depends very largely upon the viscosity of the cutting medium.

PHYSICAL AND CHEMICAL POINTS OF INTEREST.

4. *Cutting Oils.*—The mineral oils, which are best suited to be used as cutting lubricants, either alone or mixed with animal or vegetable oil, are mineral oils, preferably of pale color, of low viscosity, ranging from 100 secs. to 200 secs. Redwood at 100 degrees F. The lower viscosity oils may be used for high-speed conditions, and oils with higher viscosity may be used for slow-speed conditions. Of the animal oils used either alone or in admixture, tinged lard oil containing as much as 10 to 15 per cent of free fatty acid is most frequently employed. Prime lard oil is almost free from acid; it is much more expensive than tinged lard oil, but is less inclined to gum under severe conditions (heavy cut and high speed).

Lard oil congeals in cold weather, so that, wherever possible, a mixture of lard oil and low cold-test mineral oil is to be preferred on account of greater fluidity in the cold. Experience has proved that most cutting lubricants containing vegetable oils, particularly if they are heavily blown (*i.e.,* thickened by oxidation), are liable to produce gummy deposits in circulation systems. These deposits interfere with the proper operation of the machine, and necessitate frequent cleaning of the machines, which not only increases the costs, but also decreases the output.

Cottonseed oil oxidizes more readily than rape oil, and should not be used in the manufacture of cutting lubricants that are to be used in a circulation system. Animal oils are not so easily oxidized in a circulation system as are vegetable oils.

Experience shows that lard oil only produces deposits in circulation systems under severe operative conditions when the percentage of free fatty acids exceeds, say, 10 per cent.

When oils of low volatility are used a certain proportion of the oil is evaporated by the heat produced in the work; this is objectionable on account of the smoke and fumes created. Where cutting emulsions are used steam only is produced.

Cutting oils are nearly always used "straight"—*i.e.,* without admixture of water.

Certain cutting oils containing at least 5 per cent of free fatty acid and preferably more than 20 per cent of saponifiable oil (animal or vegetable oil) may either be used "straight" or in the form of cutting emulsions. They will emulsify with water to which the requisite amount of alkali (soda ash, borax, etc.) has been added.

Soluble Oils and Soluble Compounds.—Soluble oils are prepared by dissolving a soap (usually less than 20 per cent) in a mixture of mineral oil (usually less than 70 per cent) and saponifiable oil (usually more than 15 per cent).

The saponifiable oils used in making the soap are either of animal or vegetable origin, such as lard oil or other olein from animal fat, whale oil, wool grease, castor oil, sulphonated castor oil, rape oil, cottonseed oil, resin, resin oil, etc., and the oils are saponified by means of caustic soda or caustic potash.

Some soluble oils or compounds contain a small percentage of alcohol or ammonia, which causes the emulsion to form more readily.

When soluble oils containing ammonia are stored for several months in wooden barrels, the ammonia is sometimes absorbed by the barrel, and the oil is no longer completely soluble in water.

With cutting emulsions made from soluble oils containing ammonia, the ammonia volatilizes under severe conditions of service, and a scum is produced on the surface of the emulsion which is objectionable, as it tends to clog the pipes in the circulation system, and by adhering to the swarf or chips causes considerable loss of oil. Soluble oils containing a large percentage of resin or resin oil have a tendency to cause gumming.

Soluble compounds are made on similar lines to soluble oils, except that they contain 10 to 50 per cent of water and are in a semi-solid and semi-emulsified condition. They are not so easily mixed with water as are soluble oils, and for this reason the latter are usually preferred.

PART II.—SKIN DISEASES PRODUCED BY LUBRICANTS.

1. Oil rashes are, generally speaking, of two kinds—the first is due to plugging of the small glands at the root of the hairs on the arms and legs of workers, the second to mechanical injury to the skin produced by metallic particles suspended in the cutting lubricant.

(*a*) *Plugging of the Glands of the Hair Follicles.*

Primarily this is purely mechanical; a mixture of oil and dirt blocks the minute openings of these glands and sets up inflammation round the hair (folliculitis). The inflammation commenced in this way may lead on to suppuration or abscess formation (a boil). If many hairs are effected the arm presents an appearance of a crop of raised red spots (papules), with a black spot as a center, or, if the inflammation has gone as far as suppuration (abscess formation), a yellow head.

(*b*) *Mechanical Injury to the Skin by Metallic Particles.*

Minute metallic particles suspended in the cutting lubricant may produce injury to the skin. This occurs chiefly on the hands, where two surfaces are rubbed together—*e.g.*, the skin between the fingers. Injury to the skin may also be produced on any part of the hands and arms by wiping with a cloth or rag while the hands or arms are coated with a film of fluid in which metallic particles are suspended. Injury to the skin allows germs to enter and causes septic infection.

2. *Prevention.*—(*a*) *Cleanliness of the Worker.*—Washing accommodation for workers in contact with oil must be on a liberal scale. Hot water, soap, and scrubbing brushes are essential. Workers should be instructed not to wipe their hands on rags, etc., before washing, and to avoid washing their hands in the cutting compounds.

Ether soap, which dissolves oil, has been found useful in preventing inflammation of the hair follicles. Dusting the arms with a powder containing equal parts of starch and zinc oxide before commencing work prevents the action of the oil on the skin.

(*b*) *Cleanness of the Lubricant.*—Care must be taken in the handling of the constituents before blending that they have not undergone changes (*e.g.*, formation of free fatty acid).

Constant removal of metal particles is necessary to avoid injury to the

skin. Filtration, such as is provided on the machines, and centrifugal action are insufficient to remove the minute metal particles which may injure the skin. Where cutting oils (straight oils) are used, their viscosity can be diminished by heat sufficiently to allow the particles to sink without affecting their value as lubricants. This operation completely removes all metal particles. In other lubricants where such a procedure is impossible it is necessary constantly to change and renew the cutting lubricants.

(*c*) *Cleanness of the Machines.*—Frequent cleaning of the machines with the removal of all the old lubricant from all parts of the machine is essential.

3. *Addition of Disinfectants or Antiseptics to the Lubricants.*—Various antiseptics, carbolic acid (1 to 2 per cent) being the most common, have been added to the lubricant to prevent rashes, and in the case of cutting emulsions 0.5 per cent of disinfectants soluble in water have been used for this purpose. The results obtained have not been altogether satisfactory, and reliance cannot be placed upon such a method to prevent skin rashes.

4. *Sterilization by Heat.*—It has been suggested to heat the cutting oil to 300 degrees F. for a short period with a view to sterilizing it, as well as to increase its antiseptic or germicidal action.

Laboratory experiments in America have shown that used oil possesses rather marked germicidal effects, and in view of the fact that the used oil becomes heated during use, attempts were made to determine whether heating new oil would also bestow upon it germicidal powers. Apparently, heating does produce such a change, but the temperature required is upwards of 125 degrees C. The actual temperature required to produce this germicidal action in the oil has not yet been determined, but it has been recommended to mix new oil with the used oil before filtering and heating, so that the new oil would possess to some extent the germicidal power of the used oil.

5. *Removal of Workers with Septic Infection of the Hands.*—Workers whose hands become the seat of septic infection should not be allowed to work on machines, as they are liable to infect the oil with germs and so infect others.

6. *Treatment.*—(*a*) *Folliculitis Produced by Blocking of the Glands.*—As a general rule, frequent washing with soap and hot water is sufficient to produce a rapid cure. The skin may be subsequently dusted with zinc oxide and starch powder.

It has been found that where this is insufficient a mild antiseptic applied on lint has relieved the irritation and given good results.

(*b*) *Septic Infection of the Skin Due to Cuts.*—Septic infection should be treated on general principles by the application of suitable antiseptic dressings.

7. *Susceptibility.*—Certain individuals appear to be particularly susceptible to the action of lubricants. Such persons when found should be removed from contact with oil.—"Mechanical World."

THE PROPULSION OF CARGO SHIPS WITH PARALLEL MIDDLE BODY.

*Variations of Shaft Horsepower, Propeller Revolutions, and Propulsive Coefficient with Longitudinal Position of the Parallel Middle Body in a Single-Screw Cargo Ship.**

BY NAVAL CONSTRUCTOR WILLIAM MCENTEE, U. S. N., MEMBER.

Previous experiments made at the United States Experimental Model Basin indicate† that for slow ships of full lines, such as are used for the ordinary single-screw cargo ship, it is desirable to use a certain amount of parallel middle body. In those experiments the different models used had varying lengths of parallel middle body distributed equally forward and abaft the midship section. The present investigation had for its object the determination of the best fore and aft position for the parallel part of the ship, and the investigation of the effect of this variation on the shaft horsepower, propulsive coefficient, and wake and thrust deduction factors.

Four 20-foot models were made corresponding to ships of 400 feet length between perpendiculars and of 57.3 feet beam, 26 feet draught, and 13,137 tons displacement when fully loaded. A longitudinal or prismatic coefficient of 0.788 was chosen as representing about the present practice in cargo carriers of this type. A constant length of the parallel middle body was taken equal to ·33 per cent of the ship's length. This percentage was found in the investigations referred to above to give about the minimum residuary resistance for the speeds attained in practice for ships of this type.

In Fig. 1, annexed, are shown the lines of model 2,023 which were used as the parent form. In Fig. 2, are shown the curves of sectional area for the four models and the parent form. Model 2,132 was made very full at the entrance, with a fine run. Model 2,135 was made relatively very fine at the entrance and full in the run. Models 2,133 and 2,134 were intermediate between the two extreme models. The model with the very full bow and that with the very full stern were purposely made of an extreme type beyond anything that might be expected to be used, in order to give a wide scope to the investigation and to insure that the limiting condition as regards power should be obtained at either extreme.

In the four models the middle section of the parallel middle body—that is to say, the section which divided the parallel middle body into two equal portions—was placed at varying distances from the forward perpendicular, amounting to 31.3 per cent, 38.5 per cent, 53.4 per cent and 60.5 per cent of the length of the ship, respectively.

The models were carefully made and all were fitted with the same cast stern frame, which included the stern bearing for the propeller shaft. The stern frame had the rudder cast with it. The whole frame and rudder was fitted to each of the four models before the self-propulsion experiments were undertaken, and, together with the propeller shaft, propeller and dynamometer were transferred from one model to the other as the experiments with each model were completed.

The dynamometer consisted of a small direct-current motor, the armature shaft of which was directly connected with the propeller shaft by means of a flexible coupling. The armature shaft was free to float fore and aft and aft in its bearings about 7/16 inch in an axial direction. The armature shaft was connected to a calibrated spring by means of a thrust

*Paper read before the Society of Naval Architects and Marine Engineers, at Philadelphia, November, 1918.
†"Speed and Power of Ships," by D. W. Taylor.

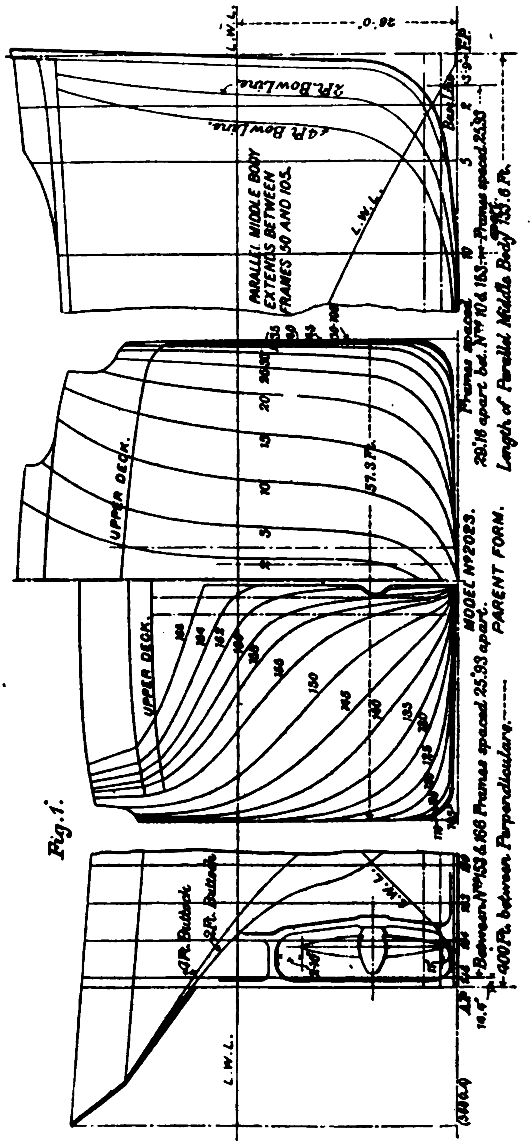

PARALLEL MIDDLE BODY
EXTENDS BETWEEN
FRAMES 50 AND 105.
UPPER DECK.
MODEL No 2023.
PARENT FORM.
L.W.L.

Fig. 1.

bearing, so that the axial displacement of the armature shaft gave a measurement of the propeller thrusts. Similarly, the frame of the motor was mounted so as to rotate in independent bearings. The torque developed by the motor acted against a calibrated spring so that the deflec-

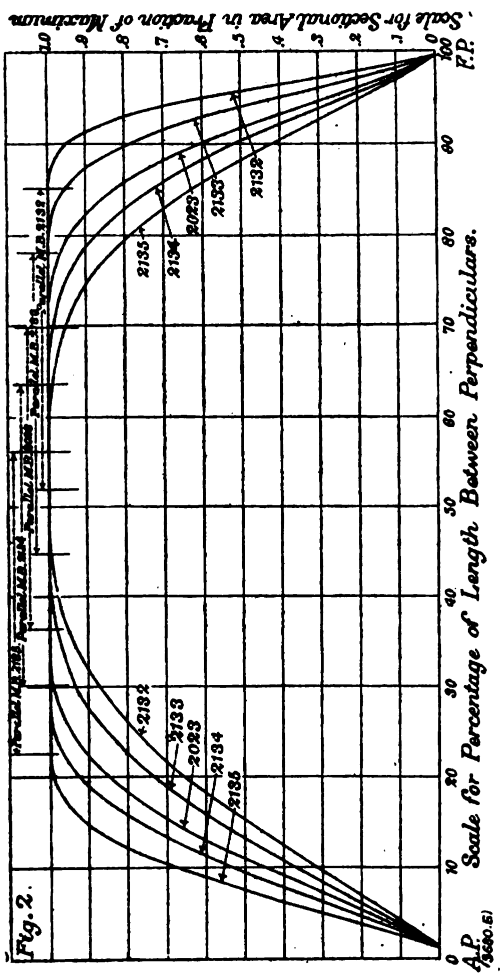


Fig. 2.

tion of the spring indicated the torque of the motor. In addition to this there were suitable means provided for measuring the revolutions of the shaft.

The order of procedure in making the tests was as follows: The shaft and dynamometer were carefully lined up and the whole run for a sufficient time to warm up the bearings and reduce the bearing friction as much as possible. Owing to the fact that the dynamometer was placed

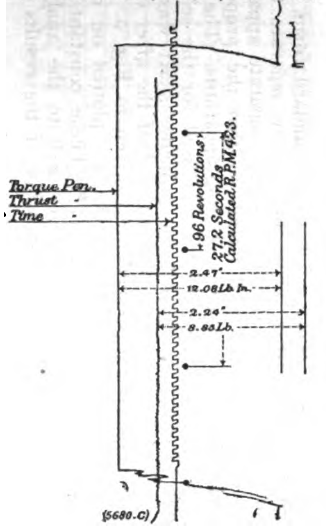
very close to the stern, but a short length of propeller shafting was necessary, and this was supported by two self-aligning bearings, one at the stern bearing and the other at the forward end of the stern tube. With the propeller shaft in place and everything working freely the model was towed in the Model Basin beneath the towing carriage at several different speeds, and the propeller shaft, without propeller, run at the range of revolutions to be covered in the course of the experiments. The propeller was then fitted to the shaft and cards for torque and thrust and revolutions per minute were taken with the model self-propelled at different speeds. In these tests the model was guided by two plates about 10 inches in width placed at either end of the model, so as to steer it in a straight course. The guide plates floated between the guiding points attached to the carriage, but the towing carriage did not exercise any force on the model in a fore-and-aft direction. Starting at low speeds corresponding to about 5 knots for the ship, the towing carriage was adjusted to run at a uniform speed. The rheostat controlling the speed of the propeller dynamometer was then adjusted so that the thrust of the propeller would just keep the model running as fast as the towing carriage, without striking the stops, which were placed at an interval of 6 inches. Thus, starting with the model in the mid position, it was free to gain or lose a distance of 3 inches as compared with the towing carriage before striking either stop. When the propeller was running at the proper speed to keep the model up with the towing carriage the record of thrust, torque and revolutions per minute was taken. If, in the course of the run, the model struck either stop on the carriage the run was discarded and another run made. Having obtained the desired data at the lowest speed, the carriage speed was increased for subsequent runs and similar data taken at higher speeds. The range covered corresponded to speeds of 5 knots to 12½ knots for the ship. About 40 different runs were made with each model, giving a corresponding number of points for plotting the torque, thrust and revolutions per minute curves.

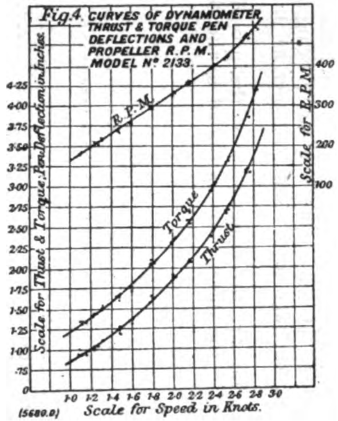
The armature of the propeller dynamometer was especially designed to reduce to a minimum the amount of magnetic thrust. This thrust increased with the torque and amounted to 0.17 pound when the armature was displaced 7/16 inch and the torque delivered to the shaft was 16 pound-inch. Neglecting this at higher powers would have caused an error in thrust measurements of 1.4 per cent, but would not have caused any error in the power measurements. However, this magnetic thrust was separately calibrated, and corrections for it were made in working up the results of the experiments.

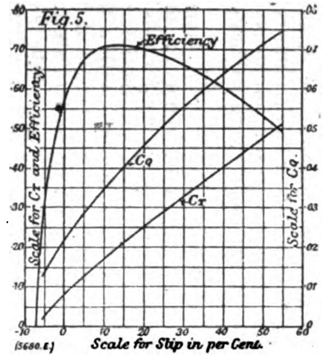
It is interesting to note that experiments of this kind with a very heavy model, in this case a displacement of 3,774 pounds, afford a very sensitive means of checking the uniformity of speed of the towing carriage. If the towing carriage itself were not very carefully adjusted, so as to eliminate slight variations in power required to drive it, owing to variations in the level of the tracks or uneven friction of the driving wheels or guide wheels, the resulting small accelerations and retardations of the carriage were made very apparent. As the model, driven by its own propeller, when supplied with a constant voltage, ran at very uniform speeds, it was possible to see the towing carriage gaining or losing distance of a few inches in 5 seconds or 10 seconds, as slight inequalities of speed due to small variations in the resistance of the driving mechanism developed. In order to conduct self-propulsion experiments successfully with the method followed, it was necessary to have the towing carriage running in excellent condition.

Immediately after completion of the self-propulsion tests on the model the propeller was removed and the runs to obtain the shaft friction and

Fig. 3. DYNAMOMETER CARD SHOWING PROPELLER THRUST, TORQUE, AND R.P.M.







Curves of Propeller Characteristics.
Propeller No. 224.

Pitch ratio	.880
Ratio projected area to disc area	.266
Mean width ratio	.200
Blade thickness fraction	.040

$$C_Q = \frac{Q}{n^2 p^2 d^3} \qquad C_T = \frac{T}{n^2 p^2 d^2},$$

d = diameter in feet.
p = pitch in feet.
n = revolutions per second.
T = thrust in pounds.
Q = torque in pound-feet.

the thrust without propeller were repeated. The model was next connected with the resistance dynamometer on the towing carriage and the usual model resistance data taken. This insured that the conditions of test both for self-propulsion and for the resistance of the model would be uniform as regards conditions of the model, temperature of water, etc. Experiments on the four models were run on succeeding days, and the springs of the propeller dynamometer were calibrated both before and after the tests.

In Fig. 3 is shown one of the dynamometer cards taken for a model running at a speed corresponding to about 10½ knots for the ship. It will be noted that the deflection of the torque spring for this condition was 2.47 inches, for the thrust spring 2.24 inches, and the interval over which the revolutions were measured was 1.78 inches. With a little practice it was possible to read both the thrust and the torque deflections within 1/100 of an inch. The revolutions could be read to an accuracy of less than one-half of 1 per cent. Considering that the data taken are plotted and the whole averaged by means of faired curves, it is estimated that the results obtained as regards the three elements measured are correct within 1 per cent.

In Fig. 4 are shown the actual observations of the torque and thrust pen deflections, and the revolutions per minute of the propeller shaft plotted on speed of the model, the latter being measured by the speed of the towing carriage in the same manner as in the ordinary resistance tests.

The following are the dimensions of the propeller used in the experiments and also the dimensions expanded to the ship scale:

	Model. In.	Ship. Ft.	Ship. In.
Diameter	10.125	16	7
Pitch	9.0	14	9
Pitch ratio	0.889		
Mean width ratio	0.20		
Number of blades	3.		
Ratio of projected to disc area	0.266		
Blade thickness fraction	0.04		

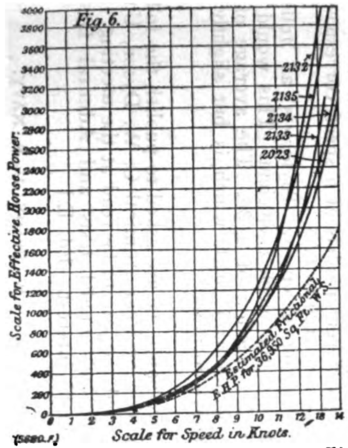
The propeller had three blades of Taylor's standard form.

The propeller characteristics were obtained by separate tests of the propeller model run in free water, that is in a separate apparatus where the propeller shaft projected well ahead, so that the propeller ran in water undisturbed by the action of the testing apparatus. The same motor dynamometer was used for the tests as was used for the self-propulsion tests, the only difference being that the propeller shaft was coupled to the forward end of the armature shaft instead of the after end.

The characteristics of the propeller are given in Fig. 5. The thrust constant C_T, and the torque constant C_Q, are plotted on nominal slip following the method used by Schaffran.* These constants, which are in non-dimensional form, lend themselves well to the analysis of self propulsion experiments and to the extension of the results to the full-size ship.

The results of the investigation are given in Figs. 6, 7 and 8. An examination of the effective horsepower curves and the shaft horsepower

* "Systematische Propellerversuche," K. Schaffran, Schiffbau, September 22, 1915.

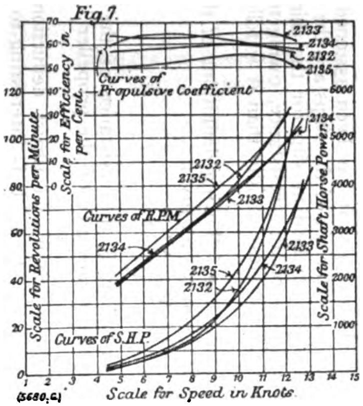


(5680.F)

Curves of Effective Horse Power for 400-Ft. Cargo Ship.

With variable location of parallel middle body.
Beam = 57·3 ft., draft = 26 ft., displacement = 13,137 tons.
Length of parallel middle body = 133·6 ft. or 33·4 per cent. of length.

Model No.	Centre of Parallel Middle Body from F.P. in p.c. of Length.	Length of Entrance in Beam.	Length of Run in Beam.	Appendages.	Trim.
2132	31·30 p.c.	1.02	3·51	Stern frame and rudder	E.K.
2133	33·46 p.c.	1·52	3. 13	Ditto	E.K.
2023	46·84 p.c.	2. 10	2·54	Ditto	E.K.
2134	53. 33 p.c.	2. 56	2. 00	Ditto	E.K.
2135	60·50 p.c.	3. 06	1. 59	Ditto	R.K.

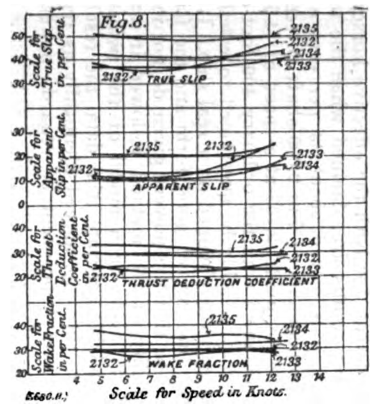


(5680.G)

Curves of Shaft Horse Power, R.P.M., and Propulsive Coefficient for 400-Ft. Cargo Ship.

With variable location of parallel middle body.
Breadth = 57·3 ft., draft = 26 ft., displacement = 13,137 tons.
Length of parallel middle body = 133·6 ft. or 33·4 per cent. of total length.

Model No.	Centre of Parallel Middle Body from F.P. in p.c. of Length.	Length of Entrance in Beams.	Length of Run in Beams.	Appendages.	Trim.
2132	31·30 p.c.	1·02	3·63	Stern frame and Rudder.	E.K.
2133	33·46 p.c.	1·52	3·13	Ditto	E.K.
2134	53·33 p.c.	2·56	2·00	Ditto	E.K.
2135	60·50 p.c.	3.06	1·51	Ditto	E.K.



(5680.H.)

Curves of Wake Fraction, Thrust Deduction Coefficient, True Slip and Apparent Slip for 400-Ft. Cargo Ship.

Variable location of parallel middle body.
Beam = 57·3 ft., draft = 26 ft., displacement = 13,137 tons.
Length of parallel middle body = 133·6 ft. or 33·4 per cent. of length.

Model No.	Centre of Parallel Middle Body from F.P. in p.c. of Length.	Length of Entrance in Beam.	Length of Run in Beam.	Appendages.	Trim
2132	31.30 p.c.	1·02	3.63	Stern frame and rudder.	E.K.
2133	33.46 p.c.	1.52	3.13	Ditto	E.K.
2134	53.33 p.c.	2.56	2.00	Ditto	E.K.
2135	60.50 p.c.	3.06	1.61	Ditto	E.K.

curves for the different models shows, as would be expected, a wide variation in power required. By plotting cross curves of power at a speed of 11 knots it will be found that, for both the effective horsepower and the shaft horsepower, the best results are obtained when the middle section of the parallel middle body is placed about 43 per cent of the ship's length from the forward perpendicular. But 2,000 shaft horsepower is required for the ship represented by model 2,133 at a speed of 11 knots. This is considerably below that obtained with the ordinary 400-foot cargo ship of this displacement and it may possibly be questioned to whether it is not lower than could be expected to be obtained on a real ship. It must be remembered, however, that the lines are of good form and the friction of the shaft has been eliminated so that the powers given do not include shaft friction. As to the reliability of the method followed it may be stated that in similar experiments made on a model of the twin-screw collier *Jupiter,* for which are available accurate trial data, the results of the model tests, when extended to the ship, agreed identically as regards the propeller revolutions and within 1 per cent as regards power and propulsive coefficient.

In Fig. 8 are shown the curves of wake fraction, thrust deduction coefficient, apparent slip, and true slip for the ship. In extending the results of the model experiments to the full-size ships it has been assumed that the wake fraction and thrust deduction coefficient of the ships are the same as for the models. The apparent slip and the true slip for the ship are less than for the model, because of less relative resistance, owing to the fact that the frictional resistance increases less rapidly than if it followed the law of comparison.

In order to avoid confusion in terms, the following definitions of thrust deduction coefficient t, and wake fraction w, are given:

$$t = \frac{T - R}{T}; \quad w = \frac{V - V'}{V};$$

in which T is the thrust of the propeller, R the resistance of the ship, V the speed of the ship, V′ the speed of advance of the propeller in the water in which it works.

As was to be expected, the wake fraction and thrust deduction coefficient both increase with fullness of the stern. It is interesting to note that in all cases the wake fraction is considerably greater than the thrust deduction coefficient, resulting in a hull efficiency greater than unity. The wake fraction obtained for the different models at a speed corresponding to 11 knots for the ship varies from 0.29 to 0.35.

The formula for wake fraction given by Taylor is as follows: $w = -0.05 + 0.5\,b$, in which b is the block coefficient. This would give for all of these models a wake fraction of 0.336. The average of the results obtained in these experiments is 0.31, which is but slightly less than that estimated by the above formula.

[In connection with the above paper we are asked to publish the following contribution to the discussion by Rear Admiral C. W. Dyson, United States Navy, which was received too late to be read at the meeting. The agreement between Mr. McEntee's model tests and Admiral Dyson's independent estimates is of much interest.—Ed. E.]

DISCUSSION BY REAR ADMIRAL C. W. DYSON, U. S. NAVY.

Commander McEntee has requested me to analyze the propeller which he used in his experiments and to make an estimate of its performance behind the various hulls which he has used in order to ascertain the degree of agreement between the results obtained by the two methods.

ESTIMATE OF PERFORMANCE.

Shaft Horse-power.

Hull.	v.	e.h.p.	$\frac{\text{e.h.p.}}{\text{E.H.P.}}$	Z.	K.	Est. Power.	Tank Power.	Log Av	Log As.	s.	Est. Revs.	Tank Revs.
2,132	8	480	0·1415	0·89	1·27	735	750	3·48	2·78	0·09	60·4	62·0
	10	1,100	0·3243	0·51	1·27	1,762	1,800	3·48	3·01	0·1274	78·7	80·5
	12	2,580	0·7607	0·12	1·27	4,325	4,500	3·48	3·23	0·1885	101·6	109·0
2,135	8	610	0·1799	0·74	1·41	1,153	1,225	3·51	2·78	0·1379	63·75	69·0
	10	1,240	0·3656	0·455	1·41	2,220	2,250	3·51	3·01	0·1564	81·44	86·0
	12	2,500	0·7371	0·138	1·41	4,607	4,550	3·51	3·23	0·1955	102·5	109·0
2,133	8	450	0·1327	0·91	1·27	702	725	3·48	2·78	0·086	60·8	61·5
	10	925	0·2727	0·588	1·27	1,472	1,480	3·48	3·01	0·1065	76·9	77·5
	11	—	—	—	—	2,030	2,000	—	—	—	86·0	87·0
	12	1,825	0·5381	0·276	1·27	3,020	2,850	3·48	3·23	0·1316	94·9	98·5
2,134	8	490	0·1445	0·855	1·27	796	850	3·48	2·78	0·098	60·9	63·0
	10	1,000	0·2949	0·553	1·27	1,596	1,700	3·48	3·01	0·1154	77·66	79·5
	12	1,825	0·5381	0·276	1·27	3,020	3,150	3·48	3·23	0·1316	94·9	97·5
2,023	8	480	0·1415	0·89	1·27	735	No data	3·48	2·78	0·09	60·4	No data
	10	925	0·2727	0·588	1·27	1,472		3·48	3·01	0·1065	76·9	
	12	1,660	0·4895	0·326	1·27	2,679		3·48	3·23	0·1167	93·3	

The agreement between the two power columns is so close that it appears to me that Commander McEntee has developed a method for obtaining quickly and cheaply all such data as it has taken me years to collect and from the data so obtained, not only to deduce absolutely correct factors for propeller design, but also so to classify hulls along such definite lines that the performances of combined hull and propeller can be forecasted with the minimum of error. He is most heartily to be congratulated upon the results of his labours up to the present time, and I hope and trust that he has not yet reached the end of his investigations.

In order to give a clear understanding it will be necessary for me to give a brief description of the method used by me and which has been obtained by many years' study of the performances of actual propellers driving actual ships over carefully measured courses.

The form of propeller blade selected from which to derive the design or performance factors is that of which the projected area is an oval with the greatest circular width at 0.7 radius of the propeller from the center.

From these performances a series of basic curves of design have been obtained from which the performance of the propeller under these basic conditions can be obtained and the performance of the propeller under any other conditions of performance derived from this basic performance by the application of suitable factors entailed by the changed conditions.

The basic conditions are denoted as follows:

I.H.P. = Basic indicated horsepower.

S.H.P. = I.H.P. $\times$ 0.92 = Basic shaft horsepower.

P.C. = Basic propulsive coefficient with maximum hull efficiency. (Taken for total projected area ratio.)

E.H.P. = I.H.P. $\times$ P.C. = Basic effective (tow-rope) horsepower.

P.A. $\div$ D.A. = Projected area ratio (outside 0.2 diameter) of three-bladed basic propeller.

4/3 P.A. $\div$ D.A. = Projected area ratio of four-bladed screw.

⅔ P.A. $\div$ D.A. = Projected area ratio of two-bladed screw.

1 — S = 1 — Basic apparent slip (for three blades) and for fullness of after body of vessel.

I.T.$_D$ = Basic indicated thrust in pounds per square inch of disc area of the propeller.

D = Diameter of propeller in feet.

P = Pitch of propeller in feet.

T.S. = Basic tip speed (three blades) in feet, of propeller.

v = Actual speed of vessel.

e.h.p. = Effective (tow-rope) horsepower for v.

The power required to deliver e.h.p. where no cavitation exists is expressed by:

$$\text{I.H.P.}_d = \text{I.H.P.} \times K \div 10z$$

where z depends for its value upon the value $\frac{\text{e.h.p.}}{\text{E.H.P.}}$ and K is the thrust deduction factor.

The revolutions corresponding to the actual conditions of resistance are found by the following equations:

$$s = \text{Apparent slip} = S\ \frac{\text{I.H.P.}_d \times Av}{\text{I.H.P.} \times A_v}$$

and

$$R_d = \frac{v \times 101.33}{P \times (1 - s)}$$

where S is the basic apparent slip and Av and A_v are factors depending upon the values of V and v.

It should be borne in mind that slight variations in the form of blade have only slight effect upon the efficiency so long as the same projected area is retained, but do have a considerable effect upon revolutions, therefore we should be prepared to find but small differences between the estimated and tank powers if both methods are correct, but considerable variation in the revolutions as the propeller used by Commander McEntee had blades of the Taylor form, while the standard-blade form used by me is an oval.

In determining the block coefficient to use for basic apparent slip, the open water (where the propeller is not covered by the limits of the load

water plane of the ship) correction is applied for hulls 2,132, 2,133 and 2,134, directly to the standard block as ordinarily obtained for hulls of the given length, beam and displacement, while for hull 2,135, which had abnormally full after-body lines, this same correction has been applied to an increased standard block, this block being 0.80 in the first three cases and 0.85 in the latter, the corresponding open-water blocks being 0.532 and 0.61.

The work of analysis and estimate follows, the estimated and tank powers and the estimated and tank revolutions being placed in parallel columns.

HULL CHARACTERISTICS.

L.B.P. = 400 ft.
B = 57.3 ft.
H = 26 ft.
Displ. = 13,137.
Nom. B.C. = 0.7716.
B ÷ L.B.P. = 0.1433.
Stand. B.C. for K = 0.80.
Est. Stand. B.C. for K (Hull 2,135) = 0.85.
D < .7 H ∴.
Stand. S.B.C. for V = 0.532.
Stand. S.B.C. for V (Hull 2,135) = 0.61.
K = 1.27, for 2,135 = 1.41.

BASIC CONDITION OF PROPELLER.

P.A. ÷ D.A. (three blades)	0.266
D	16.583 ft.
P	14.75 ft.
T.S.	5,900
P × R	1,671.
1 − S (for S.B.C. 0.532)	0.89
(for S.B.C. 0.61)	0.90
V	14.67
	14.84
I.T.D	3.1
I.H.P.	4,880
P.C.	0.695
E.H.P.	3,392.
S.H.P.	4,490

—"Engineering."

TWO VERSUS FOUR-CYCLE INTERNAL-COMBUSTION MARINE ENGINES.

By Giovanni Chiesa, Manager of Messrs. Ansaldo San Giorgio's Works at Turin.

The relative superiority of two or four-cycle internal-combustion engines for marine purposes is one of the most debated questions at the present moment from a theoretical as well as from a practical standpoint: thus it forms daily the subject of discussions, lectures and articles in technical reviews. The chief purpose of this article is to coördinate the arguments

which have been alleged for and against both types in their best form of construction, and to endeavor to draw a conclusion after careful consideration of all points of the question.

The advantages which are usually attributed to the two-cycle engine as compared with the four-cycle type may be briefly stated as follow:

(A) The two-cycle engine develops a greater power than the four-cycle with the same number and size of the cylinders and the same number of revolutions. This advantage of the two-cycle type is due to the fact that the four-cycle type gives an impulse for each cylinder every two revolutions, whilst the two-cycle type gives an impulse each revolution; theoretically the two-cycle type should therefore develop, under the same conditions, a power double that of the four-cycle type. In practice, however, the said theoretical limit has never been reached, but at present it may be said that the power developed by a two-cycle engine is 175 per cent to 190 per cent of that of a four-cycle engine, and it may be added that whilst the mean effective pressure in the four-cycle type is about 5 kg. per cm.^{2} (71 pounds per square inch) that of the two-cycle type is practically of 4.4 kg. to 4.75 kg. per cm.^{2} (62 to 67 pounds per square inch).

This essential advantage of the two-cycle type brings as a consequence a remarkable reduction of space and weight, which may be approximately calculated in the following manner: As there is no reason that a four-cycle cylinder with its framing and driving gear (assuming the same intensity of stress of the materials) should weigh less than a two-cycle cylinder of the same size, and as the weight can be practically considered to be proportional to the volume swept by the piston, therefore, for the same power and number of revolutions, the cylinder of the two-cycle engine (175 per cent being taken as the power-ratio of the two-cycle to the four-cycle type) has a weight which is 57 per cent of that of the four-cycle engine.

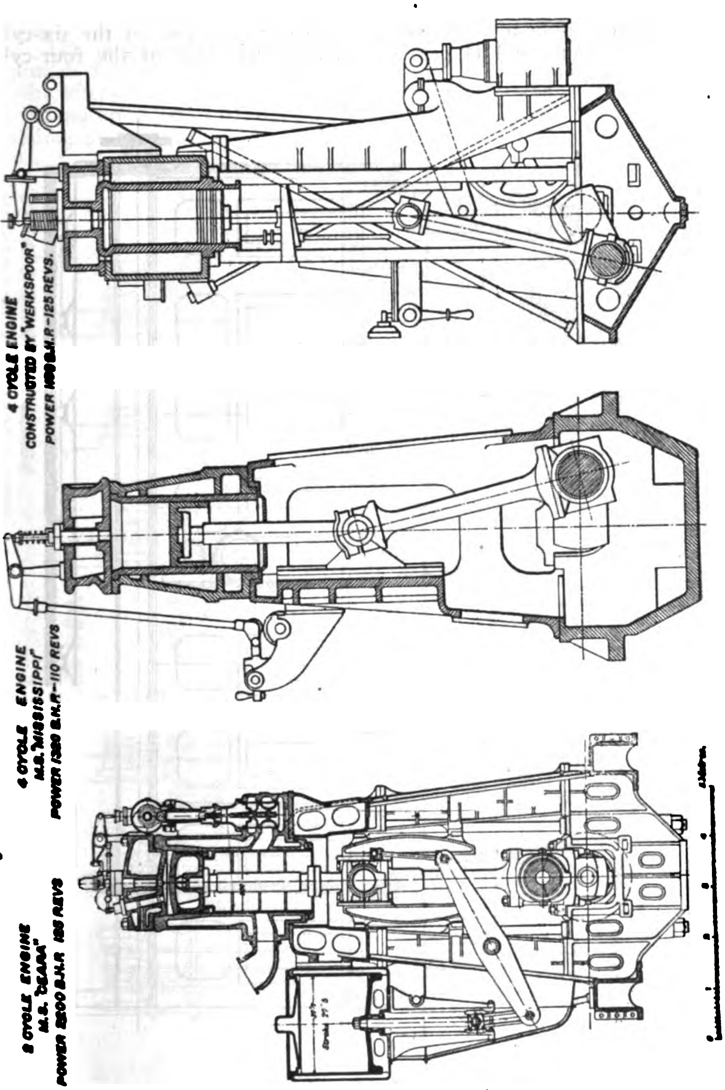
This advantage is somewhat reduced by the fact that the two-cycle engine needs scavenging pumps, and as, according to circumstances and to the different design of the pumps, their weight can be considered as being 8 per cent to 12 per cent of the weight of the cylinders, it results that the weight of the two-cycle type will be 62 per cent to 65 per cent as compared with the weight of the four-cycle. The above figures seem also practically confirmed, though there is always some difficulty in comparing numbers quoted by different constructors, for they do not always state which parts of the equipment of the plant are included or excluded from the figures published. But besides the saving of weight there is also the saving of space, and on this particular point it is preferable to refer the reader to the diagrams, Fig. 1, 2, 3 and 4, which show the two cycle engines of the ship *Ceara* constructed by the firm of Fiat San Giorgio (now Ansaldo San Giorgio), as compared with four-cycle engines of the two best-known types: Burmeister and Wain, and Werkspoor.

It must be noted that the saving in space by the two-cycle type has also as a consequence a considerable saving in the cost and weight of the engine seat as well as in the dimensions of the engine room, facilitating the supervision and control of the machines.

(B) The turning moment in the two-cycle engine is far more regular (for the same number of cylinders) than in the four-cycle type; Fig. 5 compares the diagrams of the turning moments of a four-cycle six-cylinder engine with that of a two-cycle six and four-cylinder type, illustrating the great difference in the regularity of the two types; the results of even the four-cylinder two-cycle type are far more regular than those of the six-cylinder four-cycle engine.

This advantage of the two-cycle engine is not merely theoretical, but in practice results in a minor intensity of the vibrations of the stern end of the ship, besides a reduction in size and weight of the line of shafting

Fig. 1. RELATIVE SPACES REQUIRED FOR VARIOUS TYPES.



and consequently of its fittings, such as supports, stern tube, etc. According to the rules of Lloyd's Register, the section of the shafting of a six-cylinder four-cycle engine (for the same power and the same number of revolutions) ought to be 45 per cent greater than that of the six-cylinder two-cycle engine, and 11 per cent greater than that of the four-cylinder two-cycle engine.

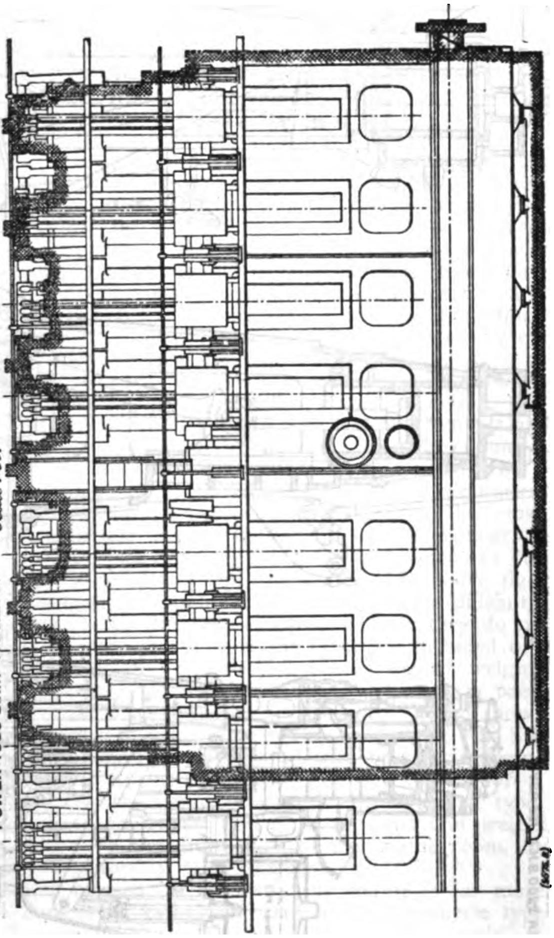


Fig. 2. RELATIVE SPACES REQUIRED.
4 CYCLE ENGINE BURMEISTER & WAIN (M.S. "SELANDIA") 2 CYCLE ENGINE (M.S. "CEARA")
POWER 1050 B.H.P.–140 REVS. *Scale 1:30.* POWER 2200 B.H.P.–125 REVS.

Furthermore, the reduced size of the flywheel in the two-cycle engine and the reduced space permits of placing the engine nearer the stern, not only saving in the length of the line of shafting but also increasing the space available on board for the cargo.

(C) The two-cycle engine offers greater facility in reversing as compared to the four-cycle type, which is due to the fact that in the former

the exhaust of the burnt gases takes place through ports in the cylinder wall, so that in order to reverse the running it is only necessary to alter the timing of the scavenging valves, of the fuel valve and of the starting valves. This alteration in the timing of the scavenging valves is very readily made by simply rotating the camshaft relatively to the crankshaft, whilst the alteration in the timing of the fuel and starting valves (these valves having but a small lift) can be readily effected by employing double cams sliding on the shaft.

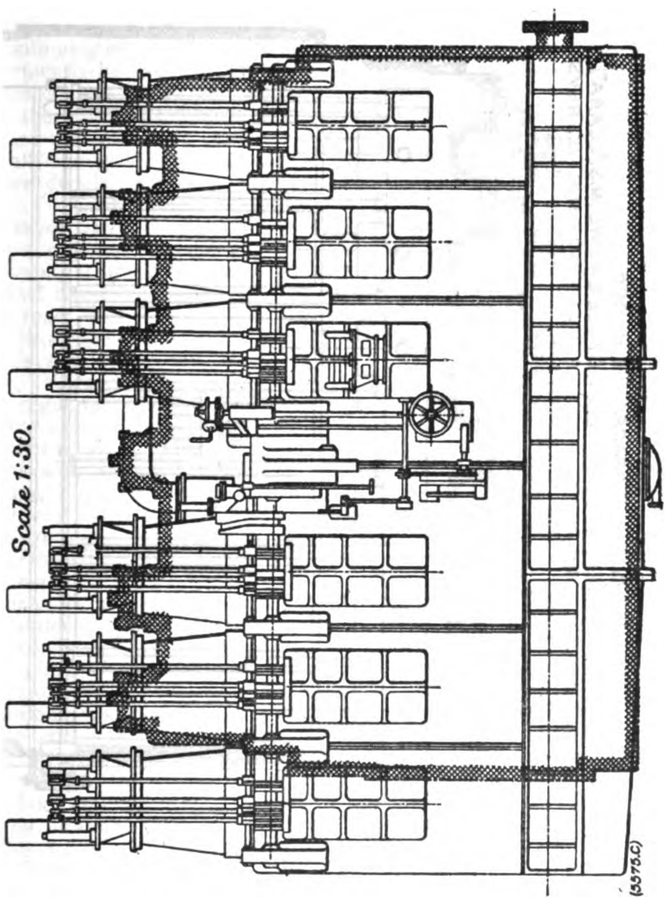


Fig. 3. RELATIVE SPACES REQUIRED.

—4 CYCLE ENGINE BURMEISTER & WAIN (M.S. MISSISSIPPI). ▬2 CYCLE ENGINE (M.S. "CEARA").

POWER 1320 B.H.P.–110 REVS. POWER 2200 B.H.P. 125 REVS.

In the four-cycle type, on the contrary, besides the alteration in the timing of the fuel and starting valves, it is necessary separately to reverse the inlet and exhaust valves; and as the latter operation requires a different rotation of the camshaft, it is not possible to employ the simple device of the two-cycle type, but much more complicated mechanisms become necessary.

Referring, further, to the starting and reversing devices, it may be added that the necessity of being able to start the engine whatever be the position in which the crankshaft has stopped, that is to have at least one of the cylinders in the inlet phase of the starting air, does not permit of a reduction in the number of the cylinders to less than six in the four-cycle type, whilst the two-cycle can be constructed with but four and keep its perfect maneuverability.

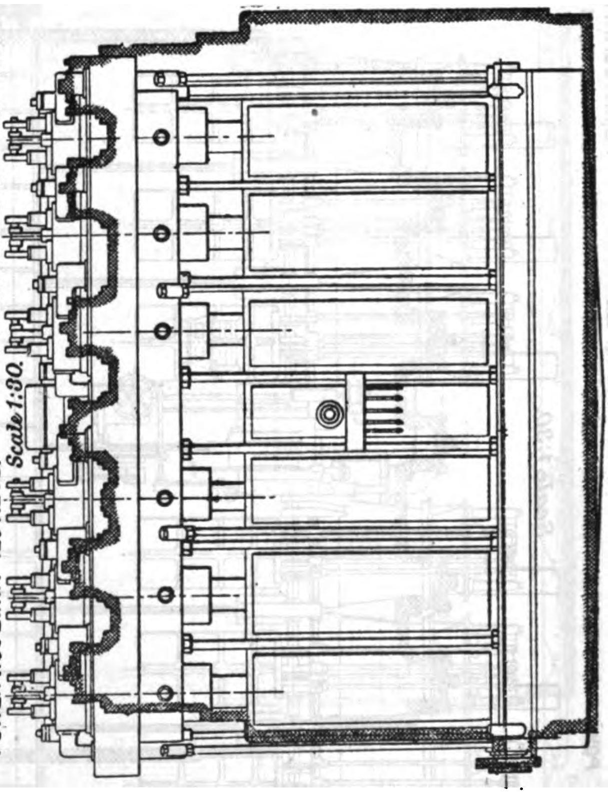


Fig. 4. RELATIVE SPACES REQUIRED.
— 4 CYCLE ENGINE "WERKSPOOR" ▬ 2 CYCLE ENGINE (M.S. "CEARA")
POWER 1100 B.H.P.—125 REVS. POWER 2200 B.H.P. 125 REVS.
Scale 1:30.

(D) With the two-cycle engine the inertia of the reciprocating parts, such as connecting rods, pistons, &c., is balanced at top dead center by the pressure on the piston, which cannot be realized in the four-cycle for the exhaust and suction strokes. As a consequence, in the four-cycle engine the piston rods are subjected to alternative compressive and tensile stresses, so that the caps and bolts of the connecting-rod heads and of the main bearings must be necessarily constructed much more strongly than in the two-cycle type in order to avoid the possibility of their breaking and the great damage which this would cause.

(E) The two-cycle engine does not require any exhaust valve for the burnt gases, and in the engine provided with port scavenging there is no need of any valve subjected to the action of the burning gases; in the four-cycle type the exhaust valves are the source of well-known troubles and even in the case where their tightness and durability is increased by using more or less complicated cooling devices, the danger of their falling into the cylinder, with all its serious consequences, can never be fully eliminated.

It should be noted that the exhaust valves in the four-cycle engine are the parts which are most sensitive to the quality of fuel and are especially liable to suffer by the asphaltum and sulphur sometimes present in heavy oils of certain origins. For a two-cycle engine without exhaust valves there may consequently be used certain kinds of fuel which are not suitable for a four-cycle engine.

Against the advantages above referred to as to the two-cycle type, the advocates of the four-cycle engine oppose some objections which partially apply to all two-cycle engines, and partially only to special types or to constructive details of them. These objections may be briefly stated as follow:

(*a*) In favor of the four-cycle type it has been said that the experience of the gas engine has led back again (after a period of preference for the two-cycle type) to the four-cycle engine; also several failures are imputed to the Diesel two-cycle engine, so that it is convenient to select again the four-cycle type.

Against this objection we may note that the example of the gas engine is not directly applicable; the two-cycle gas engine, as compared with the four-cycle, shows the disadvantage of a greater consumption and of an inefficient regulation at light loads; the greater consumption being due to the fact that a certain amount of gas is always mixed with the scavenging air because the two fluids cannot remain wholly separated, and so unburnt gas escapes with the air through the exhaust ports without producing any useful work. The bad regulation is due to the difficulty of having the right mixture in case of light loads, because in the two-cycle engine it is impossible to regulate the power without diluting the explosive mixture. Neither of the said inconveniences exist in the Diesel engines, the scavenging being made with pure air and the regulation being obtained in exactly the same manner in both the two-cycle and in four-cycle types. Moreover, it may be stated that notwithstanding the said inconveniences, which cannot be neglected, the gas two-cycle engines are still constructed, and in work for many hundred thousands of horsepower, from which we may draw the conclusion that the two-cycle engines offer other real advantages.

More suitable than the example of the gas engine for comparison is that of the hot-bulb engines where the two-cycle type is preëminent, for the Bolinder, Skandia, Fairbanks-Morse, Petter, Torbinia types, a.s.o., have almost completely eliminated the competition of the four-cycle type specially for high powers.

Referring now to some failures of the two-cycle Diesel engine, it may be said they are mainly due to constructive defects; the engines of the ship *Sebastian* have been replaced by the four-cycle type on account of the defective construction of the details of the piston cooling device; the engines of the ships *Arum* and *Arabis* have shown defective lubricating systems; numerous inconveniences have been experienced in the engines with stepped pistons, and it would therefore be wrong to attribute these failures to the type of the engine in itself instead of to defects in the design.

Other failures ought to be attributed to the inexperience of constructors who, knowing but little of the two-cycle type, have risked building engines of great power and high speed. The two-cycle type of engine is not easy

to design; the scavenging of the cylinder is a very complicated problem, and it has not been possible to study it experimentally in all details although many trials have already been made, the most important of these being that of the Krupp firm, which has made a cinematograph record by using a glass cylinder. For scavenging it is necessary that the air current in the cylinder should fully and readily displace the combustion residues without mixing with them. The study and design of a convenient system of ports passages, as well as of suitable scavenging organs, in order to obtain a perfect scavenging, and at the same time the most reduced power consumption for driving air pumps, is a problem which can only be solved completely by many trials and long and costly experiments.

Should the cylinder and the scavenging ports not be well designed and the scavenging be imperfect, the working of the engine will be bad; instead of having the cylinder filled with pure air for the combustion, there will be therein a mixture of the air with the burnt gas not fully exhausted, so that the combustion will take place irregularly and be delayed, the fuel being therefore inefficiently utilized and at the exhaust such temperatures may be reached as may greatly reduce the durability of the cylinders and of the engine itself.

The two-cycle engines which are constructed at present by experienced manufacturers, and especially those with port scavenging, are by no means less reliable than the four-cycle type. For obvious reasons it is not possible to speak of engines installed on warships (the brilliant trip of a small Russian submarine from Spezia to Arkangel cannot be forgotten), but the success of the *Monte Penedo* and of the *Ceara*, which latter is the most powerful ship in service at present, are undeniable proofs that the two-cycle engine, if well constructed, can give the best results.

Considering the extensive use made in the German navy of combustion engines of both the four-cycle and the two-cycle types, it is noteworthy that the important company, Hamburger Werft A.G., recently founded by the A.E.G., and by the Hamburg-America Linie, for the construction of motor ships, under the management of the well-known Ballin, will give the preference to the two-cycle engine.

(*b*) The supporters of the four-cycle type allege that the two-cycle engines are far more complicated, not only on account of the scavenging pumps, the piping and the receivers relating thereto, but also on account of the greater complexity of the valve gear.

Against this assertion it may be objected that the air pumps which undoubtedly constitute an added organ, by no means interfere with the reliability of the working of the engine, as they are always working at very low pressures and temperatures like the low-pressure cylinders of steam engines; and constructively it is certainly more rational to employ a suitable air pump instead of using, for half the time, for displacing the air, enormous pistons which have been designed and fitted with rings for at least a hundred times higher pressure.

Referring now to the valve gear, the complexity pertains exclusively to that two-cycle type of engine having scavenging valves in the cylinder heads, whilst in the recent type with port scavenging, besides the fuel and the starting valve (like that of the four-cycle type), there is only the scavenging valve to control. This is light and easily displaced, as it is not subjected to the highest pressures and temperatures of the cycle, and it does not require to be perfectly tight. This valve can easily be replaced by a rotary valve. In the cylinder of the four-cycle engine, instead of one scavenging valve there are two at least to be controlled, and very often two inlet and two exhaust valves, which, being placed in the combustion chamber, require to be perfectly tight and need a precise and reliable operating gear in order to withstand the effort of the powerful closing springs.

(*c*) In favor of the four-cycle type it has been furthermore affirmed that its fuel consumption is far lower than that of the two-cycle engine. Now even if it must be admitted that this objection is correct in relation to the first two-cycle engines which were constructed, and is also applicable to some present motors of defective construction, it has, nevertheless, lost much of its importance when comparing the four-cycle engine with the best-known modern two-cycle engines.

For slow two-cycle engines the consumption may be reduced under 200 grammes (0.344 pound) per B.H.P. per hour. The consumption of 194 grammes per brake horsepower has been obtained since 1915 with the 2,200 brake horsepower two-cycle engines of the ship *Ceara* on a brake test, with the most careful observation and all auxiliary pumps (such as scavenging, compressing and water, or oil pumps) directly driven, working with heavy oil of a poor quality of the density of 0.90. For the high-speed engines we may cite figures comprised between 203 grammes (0.447 pounds) and 210 grammes (0.463 pounds) per brake horsepower as resulting from the official tests of many two-cycle engines of the power of 350 brake horsepower at 480 revolutions and of 1,350 brake horsepower at 350 revolutions.

The following are the figures for consumption of fuel obtained in recent official tests on an Ansaldo San Giorgio engine of 1,300 H.P., 350 revolutions, furnished to an allied Government.

	Horse-power.	Revolu-tions.	Gr.
At 4/5 of power	1,110	292	194
At full power	1,339	352	203.1
With 10 p.c. over power	1,425	356	202.9

The consumption of lubricating oil was about 6 grammes (0.013 pounds) per horsepower per hour, although the lubrication was very abundant inasmuch as the motor was new.

The consumptions ascertained for the four-cycle engines are not much different; the data which have been published as to the slow marine four-cycle engines shew 182 grammes to 188 grammes (0.401 pounds to 0.414 pounds) per brake horsepower, often excluding the consumption of the auxiliary pumps which were separately driven. With the high-speed type the competition of 1913 of the German Admiralty for electric sets developing 300 kw. at 400 revlutions, in which the most important manufacturers specializing in the construction of the Diesel engines took part (for instance, M.A.N., Krupp, Koerting) has led to figures, which were officially published in "Der Oelmotor," 1913, and according to which the consumption in the four-cycle engines was from 197 grammes to 203 grammes (0.434 pound to 0.447 pound) per B.H.P. per hour.

It is true that some excessively low figures have been singly reported for the consumption of four-cycle engines, but they can be safely overlooked upon consideration of the circumstances of the test or of the uncommonly high consumption of the lubricating oil, which was obviously partially burnt as fuel, so that the above-stated results can be quoted as corresponding to the best up-to-date constructions. Though they still show a slight advantage for the four-cycle engine, this is not greater than 3 per cent or 5 per cent, and if we consider the other elements required for calculating the real working expenses, this difference is not of great importance. It must, indeed, be noted that the installation of two-cycle instead of four-cycle engines for a given type of ship, results in a saving in weight and space, and therefore a reduction of displacement and the possibility of increasing the run of the stern (this leading to a reduction in the power

for propelling the ship at a certain speed). This advantage amply compensates for the slightly greater fuel consumption, specially in high-speed ships where the weight and the space taken by the propelling plant have the greatest influence. Furthermore, it may be added that, even if the question of the weight and space should be regarded as a secondary one, still the two-cycle engine shows the advantage that the particulars being the same, it can develop the same power as the four-cycle one at a much lower of speed of revolution, with the consequence of a reduction in the consumption and of a much better efficiency of the propeller. It is also very likely that as the consumption in the two-cycle engines has decreased, as a consequence of rational and systematic experiments, in a few years, from 250 grammes or 260 grammes per brake horsepower, to the present values, it will still improve until it reaches and even surpasses the low consumption of the four-cycle type. Theoretically there is no reason why this should not happen, for the thermal efficiency is the same in both types, and the power required by the two-cycle engine cannot be greater than the power expended in driving the main pistons of the four-cycle engine to work half the time as pumps themselves.

Finally, besides the fuel consumption, that of the lubricating oil, which is much more expensive, ought to be considered. It is obvious that the two-cycle engine should require a less quantity of oil than the four-cycle, the load on the piston of the four-cycle engine being 50 per cent greater (with the same number of cylinders and the same ratio between diameter and stroke) than that of the two-cycle, the pressure exerted on the bearings, and on the guides being proportionally increased, so that the surfaces to be lubricated are accordingly larger. In practice, however, as the two-cycle engine may be constructed with fewer cylinders the saving in the lubricating oil is still more evident. At present the figure of 3 grammes to 4 grammes (0.00614 pounds to 0.008818 pound) per brake horsepower as the total amount of oil consumption is usually reached in high-speed engines (480 revolutions).

(*d*) As another advantage of the four-cycle type it is affirmed that the cylinder wall never reaches such high temperatures as in the two-cycle type, so that the latter are subjected to higher internal strains and thus to the danger of cracks. Now, whilst it is true that the ratio between the quantity of the fuel burnt in the four-cycle type and the surface of the combustion chamber is hardly superior to one-half of the same ratio in the two-cycle engine, other important circumstances have been overlooked which have certainly a great influence on the mean temperatures.

The action of the hot gases on the cylinder walls lasts certainly a shorter time in the two-cycle than in the four-cycle type. Whilst in the latter the cylinder walls undergo the action of the hot gases during the whole expansion and exhaust strokes, that is, practically for more than half the time, in the two-cycle engine the action of the hot gases lasts only for little more than two-thirds of the working stroke.

In two-cycle engines in which the exhaust occurs through ports, the latter open much more rapidly than the exhaust valves of the four-cycle engines, and consequently there is a much more rapid diminution in the temperature due to expansion.

Whilst the exhaust temperature in four-cycle engines is seldom below 350 degrees C. and in the high-speed engines it easily reached 450 degrees or 500 degrees C., in two-cycle engines, if well constructed, this temperature usually remains under 250 degrees C., and sometimes it only reaches 200 degrees or 210 degrees C.

Taking account of this fact a diagram has been drawn of the relative temperatures of the high-speed four and two-cycle engines (Fig. 6), which has no absolute value but merely a relative one. It is based for both types on the same hypothesis, that is, volumetric efficiency 80 per cent,

referring to air at the atmospheric pressure and at 15 degrees C., adiabatic compression from the initial temperature of 100 degrees C.; combustion at a constant pressure; adiabatic expansion, mean effective pressure 5 kg. per cm.² for the four-cycle and 4.375 kg. per cm.² for the two-cycle engine (corresponding to the ratio of 1.75 between the volumes of piston displacement); fuel consumption 198 grammes per brake horsepower for the four-cycle and 208 grammes for the two-cycle type; exhaust temperature of 450 degrees for the four-cycle engine (being the average tem-

Fig. 5. DIAGRAM COMPARING THE TURNING MOMENTS OF TWO-CYCLE AND FOUR-CYCLE ENGINES.

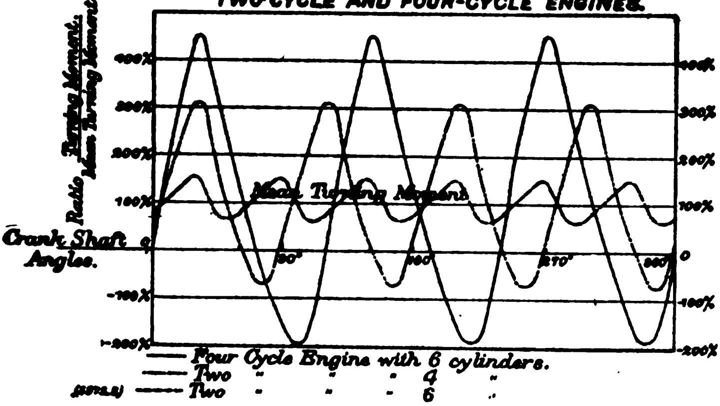
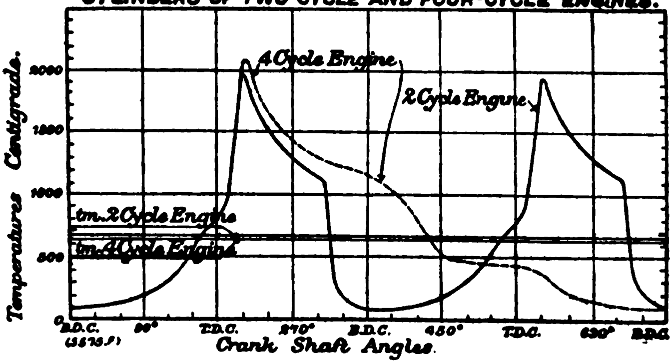


Fig. 6. DIAGRAM COMPARING THE TEMPERATURES IN THE CYLINDERS OF TWO-CYCLE AND FOUR-CYCLE ENGINES.



perature as ascertained in the high-speed engines presented at the competition of the German Admiralty above referred to). For the exhaust stroke in the four-cycle engine the curve of the temperatures is such as practically ascertained according to Dr. E. B. Wolff's experiments (see "Oelmotor," 1915), where for the suction stroke a gradual mixing of the residual exhaust gases in the compression space with the atmospheric air has been assumed.

Not one of the hypotheses above referred to is in favor of the two-cycle engine; the hypothesis of the same initial compression temperature in both types is unfavorable for the two-cycle type, as all experiments which have been made with gas engines confirm that in the two-cycle engines a much higher compression ratio can be employed than in the four-cycle, without the danger of premature ignition, and that the mixture at the beginning of the compression is therefore cooler in the two-cycle type. By measuring the diagrams by a planimeter, however, the conclusion was reached that the mean temperature of the two cycles is practically the same.

The above result is confirmed by measuring the quantities of heat absorbed by the circulating water in the two-cycle and in the four-cycle engines. Whilst in the slow four-cycle engines without piston-cooling the quantity of heat which the cooling water carries away from the cylinders is usually from 580 calories to 650 calories per brake horsepower hour, and in the high-speed engines with piston cooling from 800 calories to 850 calories (these figures have been ascertained in the four-cycle engines of the German Admiralty's competition referred to), in the two-cycle engines of a suitable construction the corresponding number of calories is from 400 to 450 in the slow and 500 to 550 in the high-speed engines. Whilst it is true that the cylinder in the two-cycle type, having the same power, exposes to the burning gases a surface which is about 30 per cent smaller than in the four-cycle, it must, however be remarked that the size of the cylinder of the four-cycle type being larger than that of the two-cycle, the thickness of the walls is consequently greater, and that a great part of the head surface in the four-cycle engine, owing to the presence of the valves, does not transmit any heat.

Taking account of all these elements it is fair to say that the two-cycle engine, from the standpoint of temperature, is in better condition than the four-cycle. The two-cycle engine, in which the inner walls of the cylinder, after the very short action of the flame, are immediately cooled by the scavenging air current (which is supplied in such quantity as to allow, besides the filling up of the cylinder, the escape of the warmest portion which entered at first) is thermally superior to the four-cycle engine in which all heat must be abstracted through the walls of the cylinders with the consequent fall of temperature in the walls and resultant internal stresses.

(*e*) The opponents of the two-cycle engine allege that in engines of this type some portion of the combustion gases remains in the cylinders, especially in the upper part of them, so that the cylinder head becomes excessively hot. Against this argument it must first be remarked that in the four-cycle engine at least 8 per cent of the burnt gases remain and fill the compression chamber when the piston has completed its exhaust stroke, and it is obvious that this remaining portion cannot but contaminate the air which is drawn in during the subsequent stroke. As regards the two-cycle engine the assertion that some residue of the burnt gases still remains in the cylinder after the scavenging operation is merely a gratuitous hypothesis, which is contradicted by the facts above referred to, according to which the quantity of heat absorbed by the walls is less in the two-cycle engine, and that in the two-cycle type the compression ratio can assume a greater value than in the four-cycle engines.

During recent and accurate tests in the test room of the Ansaldo San Giorgio works on a high-speed two-cycle engine, it has been determined that the quantity of carbon dioxide contained in gas enclosed in the cylinder during the return stroke does not exceed 0.3 per cent, and that the contents of oxygen is hardly less than the content in the pure atmospheric air, *i.e.*, 20.5 per cent instead of 20.9 per cent.

(*f*) Against the two-cycle engine it has been said that the four-cycle type can run with greater regularity than the two-cycle when working at

low speed of revolutions, owing to the fact that in the two-cycle engine the compression at low speed falls rapidly with the diminishing of the scavenging air pressure. It must, however, be noted that this observation is correct merely when it refers to two-cycle engines of bad design, in which, owing to inefficient construction, the scavenging air pressure rises, at the normal speed, to excessively high value, whilst in two-cycle engines, which have been carefully designed, even at full speed the pressure of the scavenging air remains within very small limits. By the speed reduction the pressure is also somewhat reduced, but not so as to cause failure of the ignition especially when the engine is hot. Practically, in both the two-cycle and in the four-cycle types the lowest limit of speed is dependent upon the construction of the pulverizer, and this for the two-cycle engines is more than efficient for perfect maneuvering. Moreover, it must be remarked that the turning moment of two-cycle engines being more regular, and it being possible to run with half the number of cylinders and to obtain sufficiently good regularity, the two-cycle engine shows in this particular point an advantage compared with the four-cycle type.

During official tests the above-mentioned two-cycle engine of 1,300 brake horsepower ran for a long time at 45 H.P. (*i.e.*, nearly 1/30 power) and with the corresponding speed of 115 r.p.m.

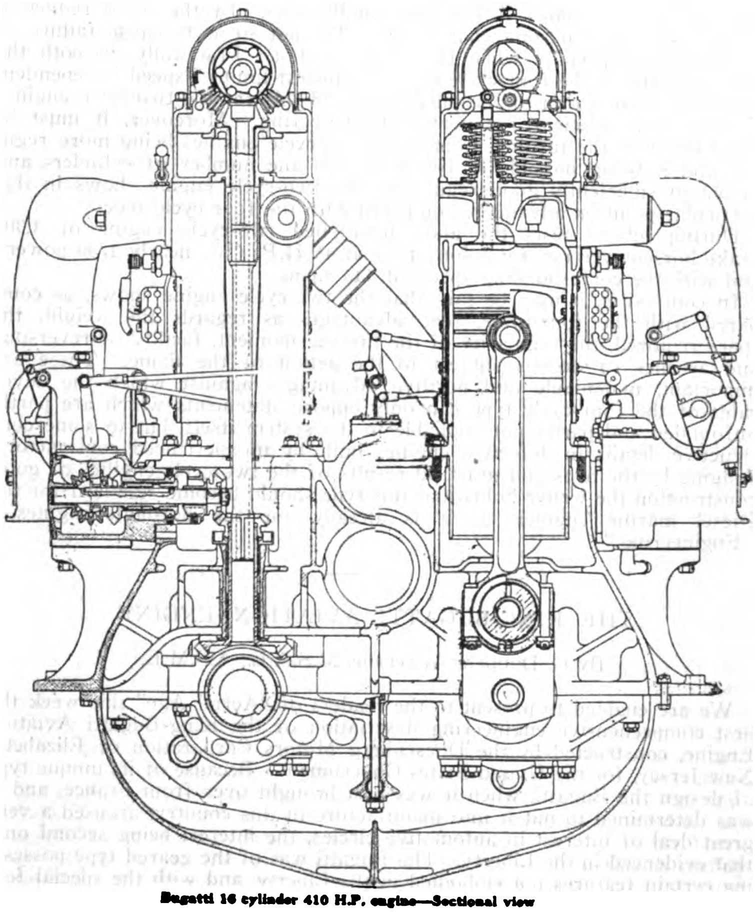
In conclusion, it may be said that the two-cycle engine shows, as compared with the four-cycle, real advantages as regards the weight, the space required, the regularity of the turning moment, facility of reversing, and absence of organs subject to the action of the flame. These are important, undeniable and positive advantages against which the advocates of the four-cycle type can only oppose statements which are partly unfounded and partly not appliable to the system itself, but to some constructive details in defective engines built by inexperienced constructors. Judging by the tests and practical results of the two-cycle engines of good construction the writer holds that this type should become standard for the Diesel marine engines, as it is already for the hot-bulb engines.—" Engineering."

THE KING-BUGATTI AVIATION ENGINE.

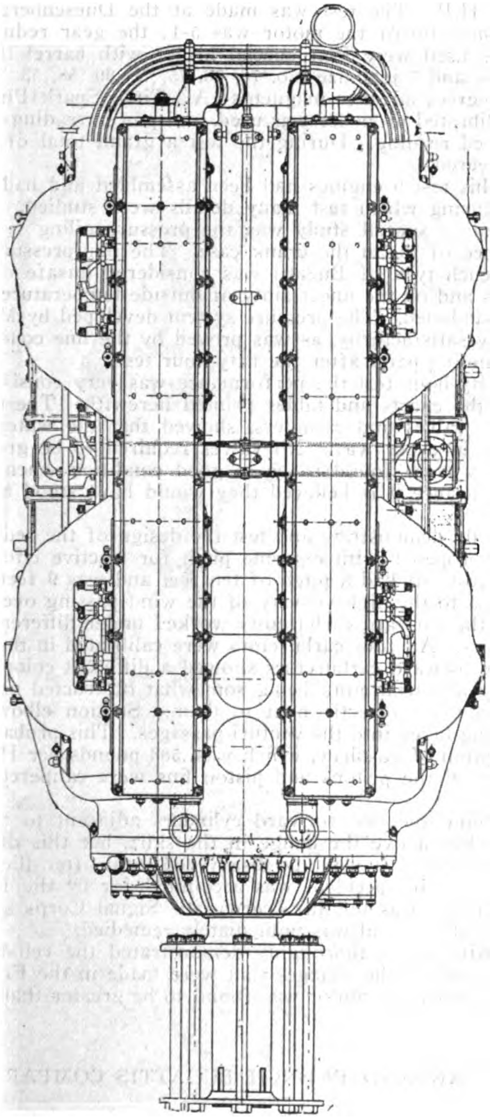
BY G. DOUGLAS WARDROP, M.S.A.E., A.S.M.E.

We are enabled to present to the readers of "Aerial Age" this week the first comprehensive engineering description of the King-Bugatti Aviation Engine, constructed by the Duesenberg Motors Corporation of Elizabeth, New Jersey, for the United States Government. Because of its unique type of design the Bugatti, when it was first brought over from France, and it was determined to put it into manufacture in this country, aroused a very great deal of interest in automotive circles, the interest being second only that evidenced in the Liberty. The Bugatti was of the geared type possessing certain features not embodied in the Liberty, and with the special feature of being able to mount a 37-millimeter cannon firing through the propeller shaft.

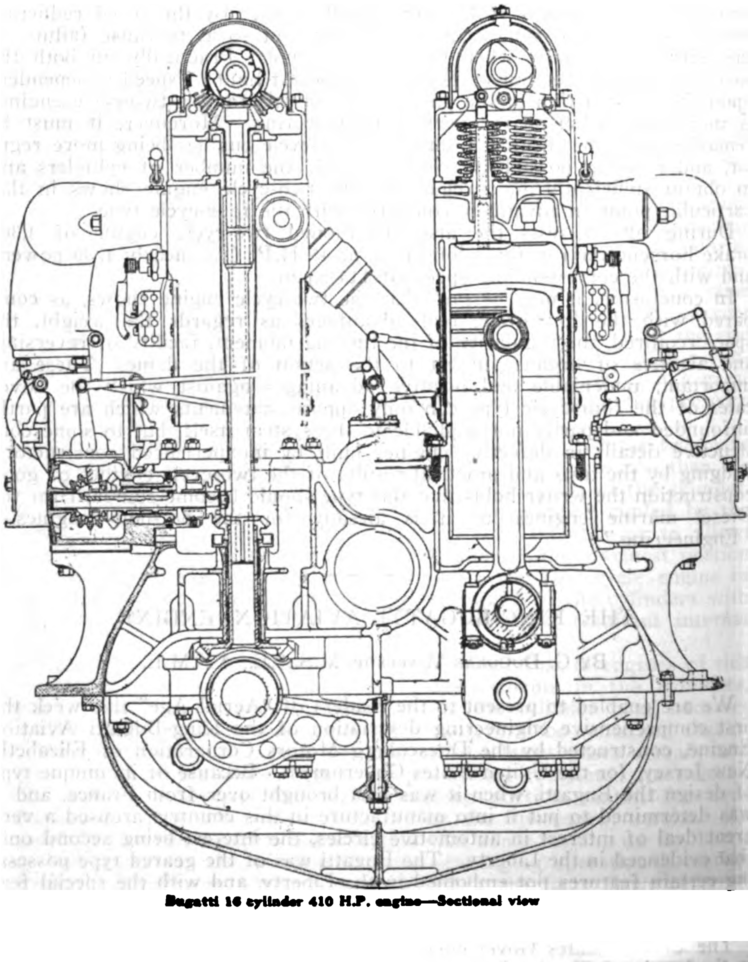
The United States Government put the redesigning of this power plant in the hands of Charles B. King, A. M. E., of the Signal Corps and he was made responsible for all changes. With a large force of draftsmen the work of adjusting the Bugatti type of design to American manufacturing methods was undertaken and upon the completion of the redesigned motor Mr. King transferred to the Aircraft Production Board a report which considered carefully all the details which are discussed hereafter and in which only minor changes have been made, since the report was handed in, which emphasizes the good judgment used when the motor was being redesigned.



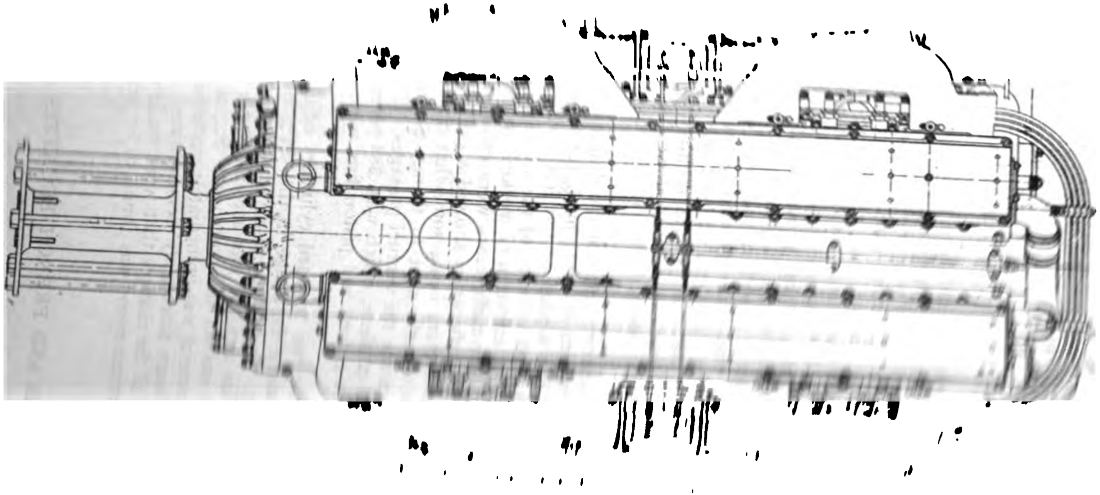
Bugatti 16 cylinder 410 H.P. engine—Sectional view



King-Bugatti 410 H.P. Engine—top view



Bugatti 16 cylinder 410 H.P. engine—Sectional view



Sixteen Cylinder King—Bugatti Aviation Engine.

The test referred to was one of fifty hours' endurance, ten periods of five hours each, the first half hour at 410 horsepower and 4 hours 30 minutes at 380 H.P. The test was made at the Duesenberg test room. The compression ratio of the motor was 5-1; the gear reduction 28 : 42. The carburetors used were four special Miller with barrel throttle, with choke 1⅞ inches and 7 jets size No. 76, 76, 75, 72, 69, 58, 53. Four Dixie Model 800-38 degrees advance magnetos; AC Titan Spark Plugs, two per cylinder. A calibrated propeller was used, and power readings used corresponding to speed reading. During the test a grand total of 19,284 H.P. hours were delivered.

Previous to this test 9 engines had been assembled and had run a total of 110 hours, during which test many details were studied. One of the points that required special study was the pressure oiling system due to cross interference of oil in the crank case. The non-pressure system as used in the French type of Bugatti was considered unsafe owing to the many long leads and on the uncertainty of outside temperature, and it was consequently abandoned. The pressure system developed by Mr. King has worked out very satisfactorily, as was proved by the fine condition of the bearings and running parts after the fifty-hour test.

During the fifty-hour test the performance was very consistent, as will be noted from the charts and tables printed herewith. The condition of the spark plugs, valves and cylinders, showed that the water circulation was satisfactory in every way. No valves required to be ground during the test and they were found to be in good condition when dissembled. From their appearance it is believed they would have stood an additional fifty hours.

In order to fully demonstrate and test the design of the reduction gears and bearings a propeller with extreme pitch for tractive effort was used throughout the test. It had a pitch of ten feet and was 9 feet 4 inches in diameter. Owing to the high velocity of the wind passing over the engine during the test the forward carburetors worked under different conditions than the rear ones. All four carburetors were calibrated in the jets before the test, but the forward carburetors showed a different color of flame at the stacks, the rear carburetors being somewhat obstructed as to air currents by the magnetos directly next to them. Suction elbows were not used, the air being taken into the venturi passages. This probably accounts for the consumption of gasolene, which was .583 pounds per H.P. hour.

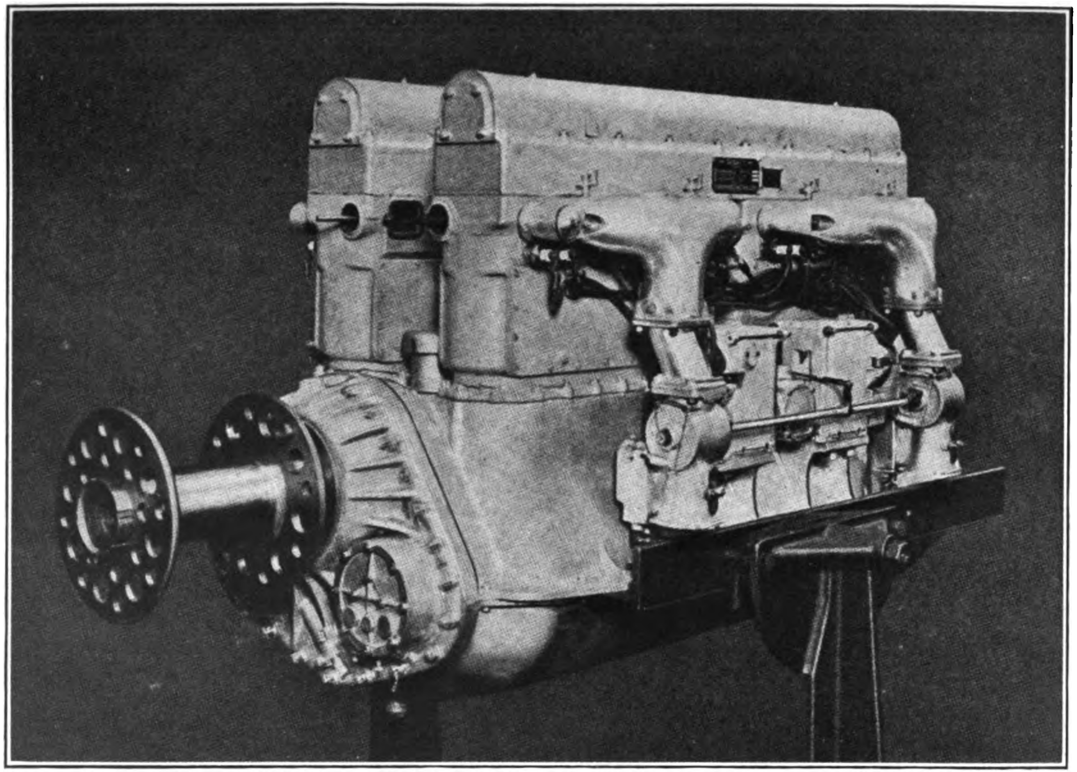
After the test all the pistons and piston fins were calipered and found round and true.

Upon dissembling the two forward cylinders adjacent to the propeller were found cracked above the flange in the skirt, but this did not interfere with the running. It was not discovered until after the engine had been taken down. This accident was accounted for by the fact that the iron of the cylinders was not in accord with Signal Corps specifications. The discrepancy of material was immediately remedied.

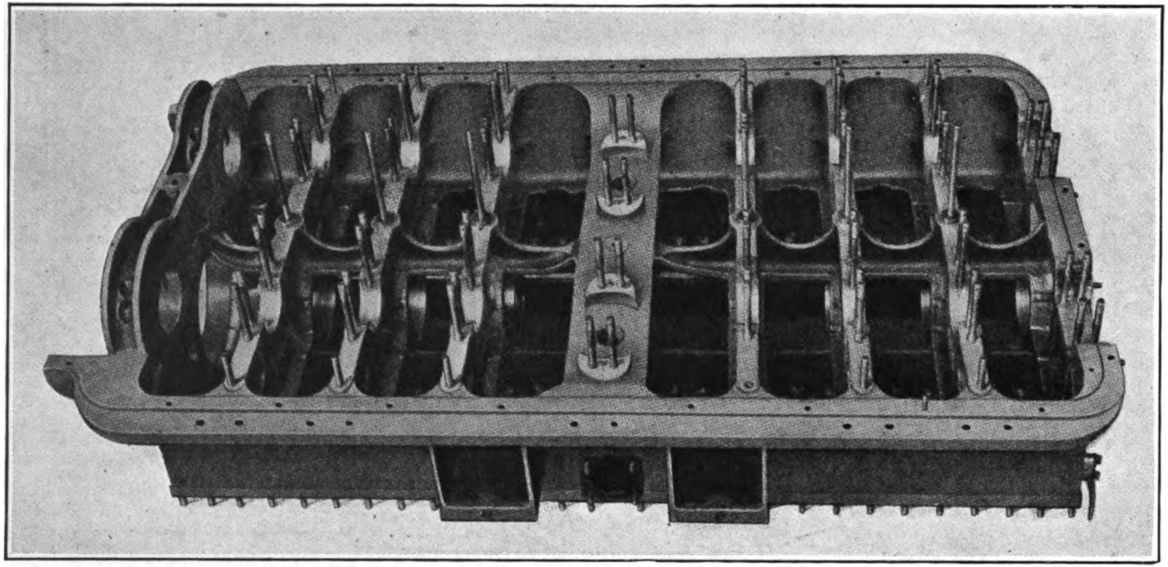
The test of fifty hours thoroughly demonstrated the reliability of the engine and endorsed all the changes that were made in the French design. The H.P. in the American motor was found to be greater than that of the French.

AMERICAN AND FRENCH BUGATTIS COMPARED.

In the course of a very interesting report Mr. King offers definite data concerning the specific changes made in the Bugatti design and the reason therefor. Owing to the fact that the French engine which was sent over to this country had had a limited test in Paris of 37 hours and had not been in flight, all of the points in the design were very carefully



Sixteen Cylinder King—Bugatti Aviation Engine.



Crank Case King—Bugatti Engine.

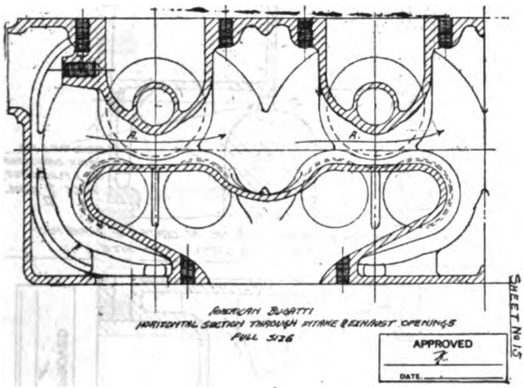

AMERICAN BUGATTI
HORIZONTAL SECTION THROUGH INTAKE & EXHAUST OPENINGS
FULL SIZE
APPROVED
DATE
SHEET No 15

Fig. 15

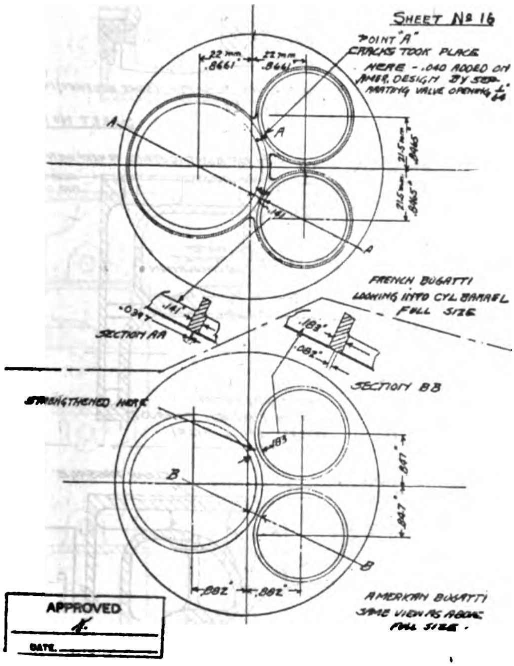

SHEET No 16
POINT "A"
CRACKS TOOK PLACE
HERE - .040 ADDED ON
AMER. DESIGN BY SEPARATING VALVE OPENING 1/64"
22 mm
.8661"
21.5 mm
.8465"
.141
.0347
SECTION AA
FRENCH BUGATTI
LOOKING INTO CYL BARREL
FULL SIZE
.183"
.082"
SECTION BB
STRENGTHENED HERE
.847"
.882"
AMERICAN BUGATTI
SAME VIEW AS ABOVE
FULL SIZE
APPROVED
DATE

Fig. 16

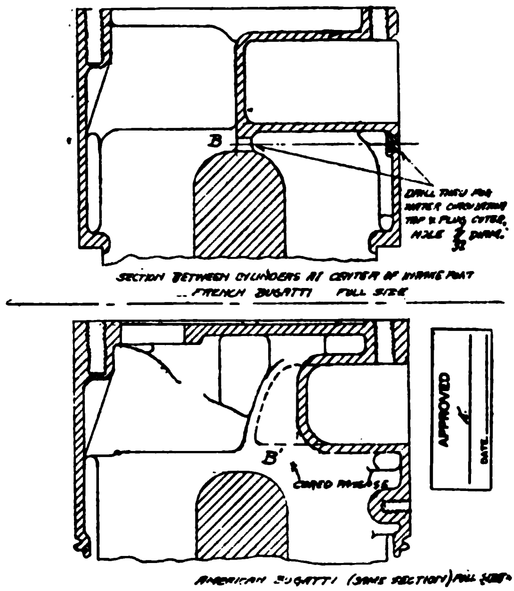

B
DRILL THRU FOR WATER CIRCULATION TAP & PLUG OUTER. HOLE 7/32 DIAM.
SECTION BETWEEN CYLINDERS AT CENTER OF INTAKE PORT
FRENCH BUGATTI FULL SIZE
B'
CORED PASSAGE
APPROVED
DATE
AMERICAN BUGATTI (SAME SECTION) FULL SIZE

Fig. 17

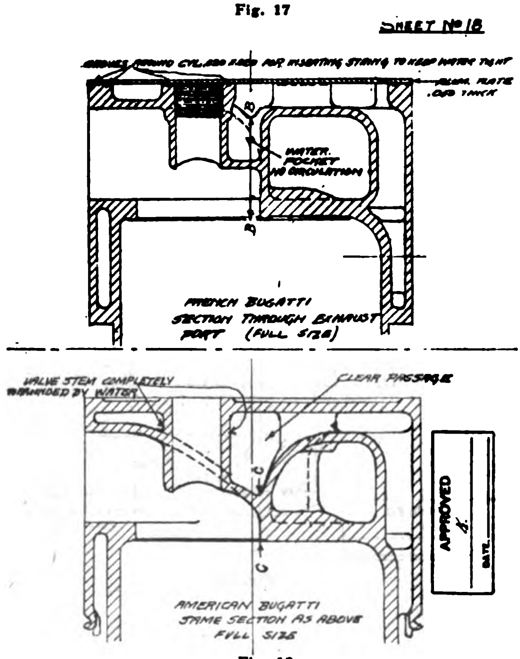

SHEET No 18
ALUM. PLATE OUR TANK
B
WATER POCKET NO CIRCULATION
B
FRENCH BUGATTI
SECTION THROUGH EXHAUST PORT (FULL SIZE)
VALVE STEM COMPLETELY SURROUNDED BY WATER
CLEAR PASSAGE
C
C
APPROVED
DATE
AMERICAN BUGATTI
SAME SECTION AS ABOVE
FULL SIZE

Fig. 18

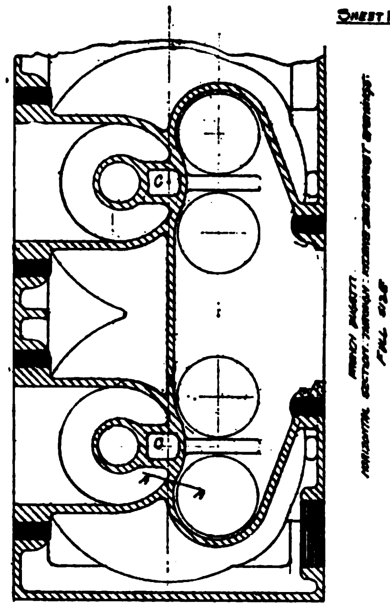

SHEET No 20
FRENCH BUGATTI
FULL SIZE
C
C

Fig. 20

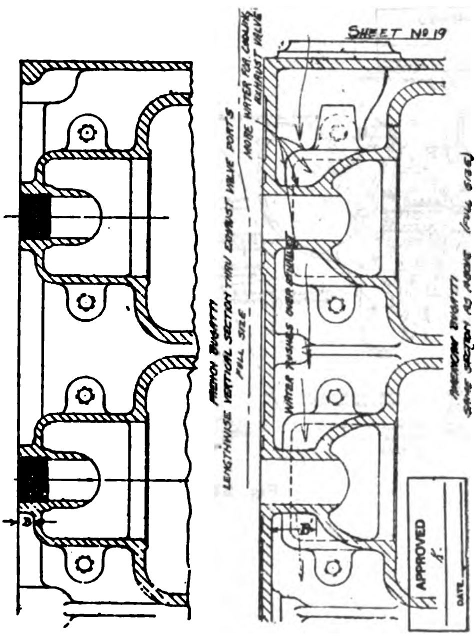

SHEET No 19
FRENCH BUGATTI
LENGTHWISE VERTICAL SECTION THRU EXHAUST VALVE PORTS
FULL SIZE
MORE WATER FOR COOLING EXHAUST VALVE
AMERICAN BUGATTI
SAME SECTION AS ABOVE (FULL SIZE)
B
B
APPROVED
DATE

Fig. 19

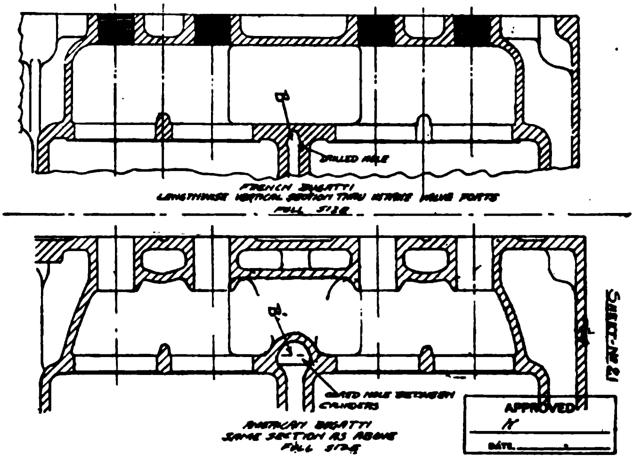

DRILLED HOLE
FRENCH BUGATTI
LENGTHWISE VERTICAL SECTION THRU INTAKE VALVE PORTS
FULL SIZE
AMERICAN BUGATTI
SAME SECTION AS ABOVE
FULL SIZE
APPROVED
DATE
SHEET No 21

Fig. 21

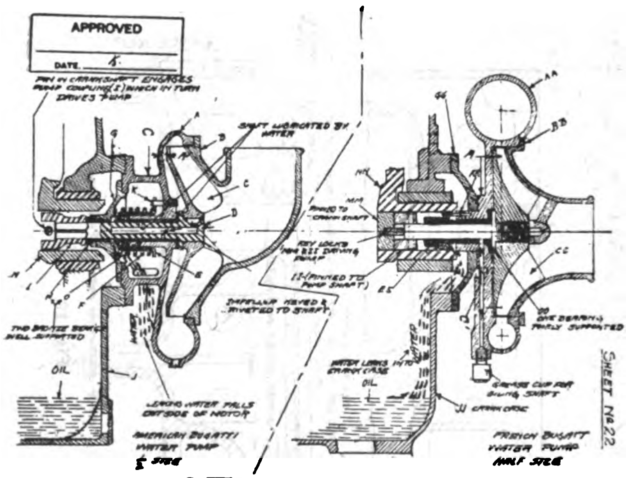

APPROVED
DATE
OIL
AMERICAN BUGATTI
WATER PUMP
FRENCH BUGATTI
WATER PUMP
HALF SIZE
SHEET No 22

Fig. 22

considered. It was soon discovered that if the job was to be made a production one numerous changes would have to be made. Some of the more important of these changes are presented in diagram.

WATER JACKETS.

It was evident that difficulties were experienced in cooling the valve seats of the French Bugatti, as cylinders and sample sections of used cylinders showed cracks between the exhaust and the inlets at Point A. See Fig. 16 showing on the plant the point of fracture. In the American design, centers of the valves were increased in both directions from center of line, namely, from 1.732 inches to 1.764 inches and also from 1.693 inches to 1.694 inches. This is clearly shown on Fig. 16. Section AA, Fig. 20, shows the deep unjacketed section of the cylinder in which the strain was set up, causing the cracked valve seats. To obviate this difficulty, the shape of the intake passage and exhaust passages were improved and the distance BV, showing the height of the unjacketed wall (see Fig. 18) was reduced to distance CC, and at the same time circulation water was swept through this formerly restricted passage. To further improve the water circulation around the valve seats the small drilled hole B (see Fig. 17 and 21) was changed to a larger cored passage B′, shown on this same Figure. The dead pocket C (see Fig. 20) was entirely eliminated and in its place clear passage was made as at D (see Fig. 15). The exhaust-valve stems were not properly taken care of as to heat transference to the water jacket. The depth of water around valve-stem guides is shown on Fig. 19, where the French Bugatti and American Bugatti are contrasted. (See B and B′.) Owing to poor conductivity through threaded portions of valve guides, the threading was eliminated and cast iron valve guides were pressed into place, thus making a much more uniformly cooled stem. The bronze guides taken from the French Bugatti showed evidence of high heat.

PROPELLER GEARS.

In transferring the French pitch of 5.6 to American pitch, it was considered that the coarser pitch, namely, 5, would be preferable (see Fig. 32), and the number of teeth was changed, but the ratio was practically maintained. It was further determined that there was an error of 4°10′ in the French Bugatti engine between the timing of the right and the left crankshafts. As this would have considerable effect in the cross firing from the magnetos and, further, the engine would not be perfectly timed, this was corrected by changing belts from eight to nine in each gear. This with a certain relation between the holes and the teeth, enables the engine to be timed as stated with a slight error of 20′. The difference in running between engines thus timed can be easily appreciated.

PROPELLER SHAFT AND BEARINGS.

At the request of Capt. Lepere the propeller was moved forward 3¾ inches to assist in stream lining the plane. This dimension would have brought an overhang of 8¾ inches forward of the front supporting bearing in the French engine. (See Figs. 23, 23A and 23B.) This was far from good practice and was not contemplated. A deep groove radial bearing was selected and the comparative overhang was reduced from 8¾ inches to 3 inches and further the gear load was taken between two such bearings. This deep groove bearing has proven in American practice to be good, as it takes both radial and thrust loads. The overhang gear as in the French engine has not been found successful in practice, as a certain wedging

action takes place from the gear load and forces the true pitch contact line to assume a diagonal line. The ball-bearing thrust as used in the French engine permits the balls to be thrown outward by centrifugal force and causes an improper contact on ball surfaces; this can be further aggravated by the wedging action of the gears. In the parts removed from the broken up French engine the thrust rings and balls were burned to a deep blue. The assembly of the propeller-shaft bearings is a difficult "in place" assembly in the French engine. In the American design the bearings are entirely an "outside" assembly. The sliding door in front cover of the French engine has been eliminated.

WATER PUMP.

The French Bugatti pump as mounted on the engine permits water leakage to enter the sump and mix with the lubricating oil. This will lead to difficulties in the operation of the engine. (See Fig. 22.) In the American model the pump was moved back, a better support added (see distances C and C′) and an opening was arranged so that the leakage could pass outside of engine sump. Two bearings were provided on pump shaft. The rotor is riveted to shaft and driven with a Woodruff key. Lubrication is provided with water which passes through center of shaft. It is found in practice in draining the system in cold weather that where the rotor is brought clear to the housing wall (see Dimension A) that water is held by capillary attraction in this narrow space and freezing results at this point and when the engine is started the pump shaft is sheared off. This space has been opened up in the American model (see Dimension A′) and the water is thus permitted to freely pass out. An elbow has been added to pump cover which can be assembled pointing in any desired direction. It is believed in this redesign that the troubles experienced during the official test have been eliminated.

CONNECTING ROD ENDS AND LINERS.

The French Bugatti engine had no liners in connecting rod ends (see Fig. 24) and depended on a thickness of babbitt of .014 inch directly on the steel shell. This being an extremely thin layer, a considerable risk was offered in a 16-cylinder engine. In case a rod was burned out steel would come in contact with steel, resulting in a seizure of rod on crank pin, a broken rod would follow, meaning the ultimate wreck of engine. In the American engine, bronze liners were put into rod ends and .047 inch of babbitt was used. In case of a burned-out rod, steel will come in contact with the bronze and the liner therefore will not seize on the crank pin. This will enable the flier to still operate his engine and make a landing. Owing to the oil pressure in the American engine, the long grooves in bearings are eliminated and in their place the short grooves are used. The replacing of liners in rods is comparatively easy and does not demand a quantity of spare connecting rods for repairs.

ROCKER ARMS.

These are shown in detail on Fig. 24A. Simplicity of machining was studied and from 14 machining operations on the French Bugatti this has been reduced to 4 on the American model. Each engine has 48 rocker arms; therefore the French engine has 672 machine operations as against 198 on the American model.

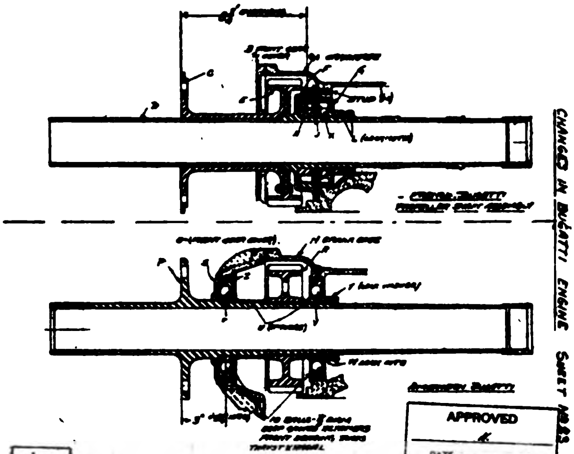


Fig. 23

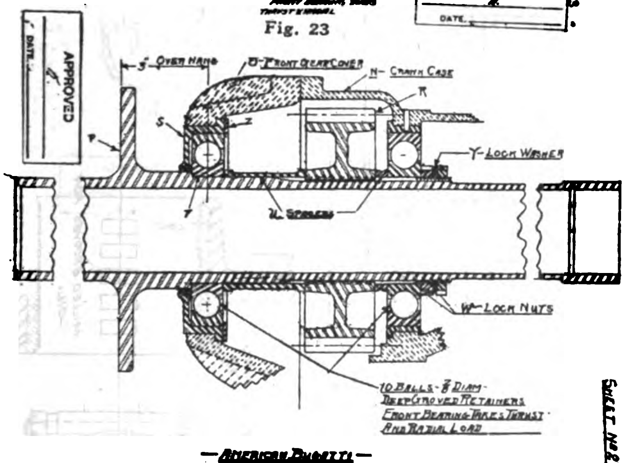


Fig. 23A

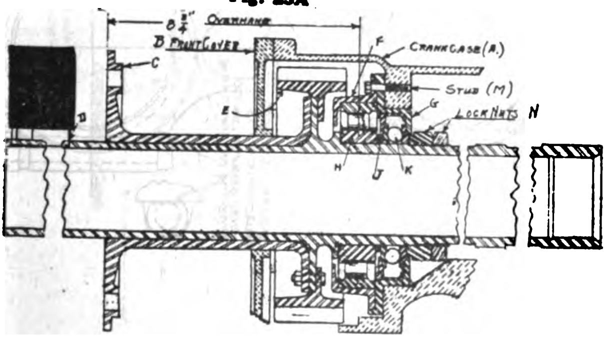


FIG. 23B

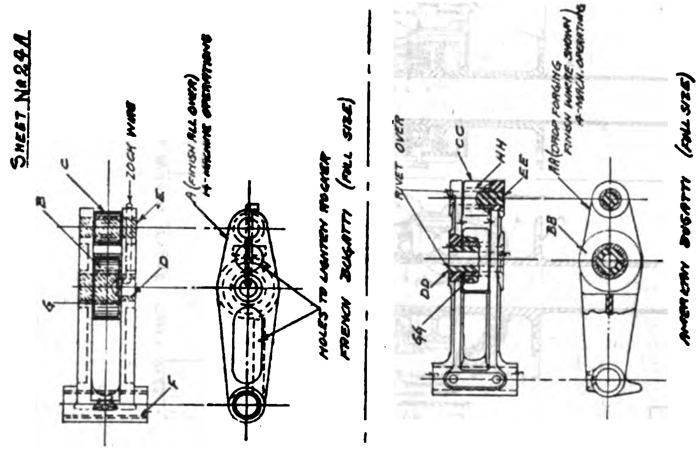


Fig. 24A

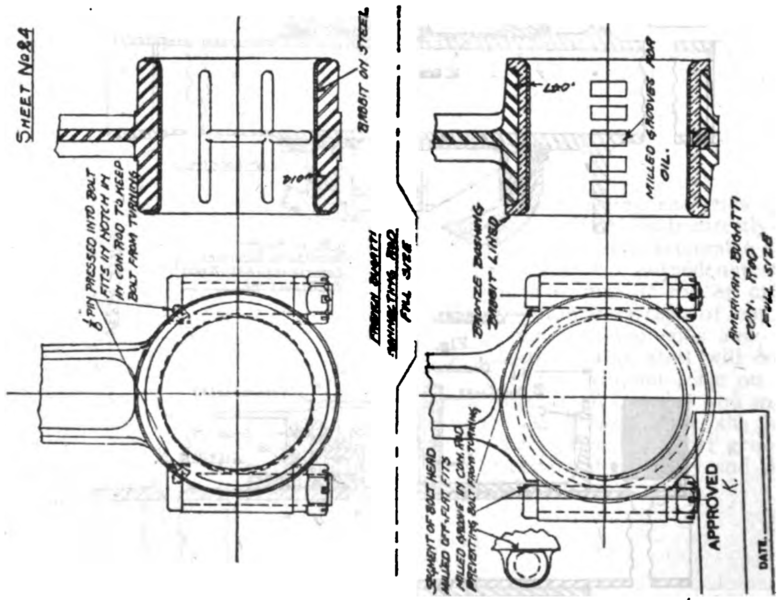


Fig. 24

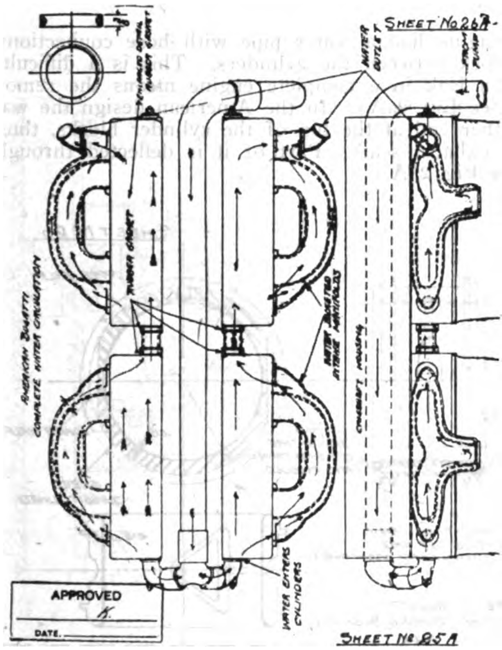


Fig. 26A

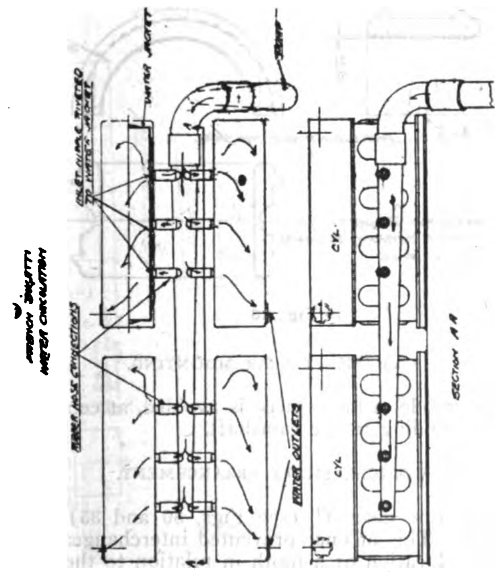


Fig. 25A

WATER CIRCULATION.

The French engine had a water pipe with hose connections, three to each cylinder block between the cylinders. This is a difficult assembly, and in case of a leak in a complete engine means the removal of the exhaust pipes (see Fig. 25A). In the American design the water can be sent through either end at the top of the cylinder blocks, thus sweeping all steam from exhaust seats. Part of it is deflected through the inlet manifolds. (See Fig. 26A.)

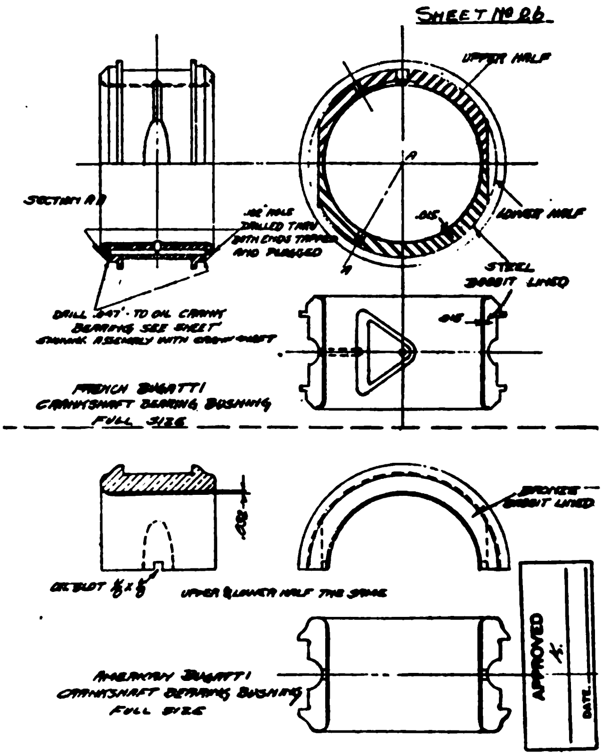


Fig. 26

CAM-SHAFT GEAR MOUNTING.

In the American model a loose piece is avoided, at center of shaft; also the small internal grinding in the camshaft.

VERTICAL SHAFT ARRANGEMENT.

In the French engine gear Y′ (see Figs. 30 and 35) was screwed on the end of shaft Z′. This at once prevented interchangeability, as it in no way determined the location of a tooth in relation to the crank pin. This in each individual French engine was to be corrected by proper timing

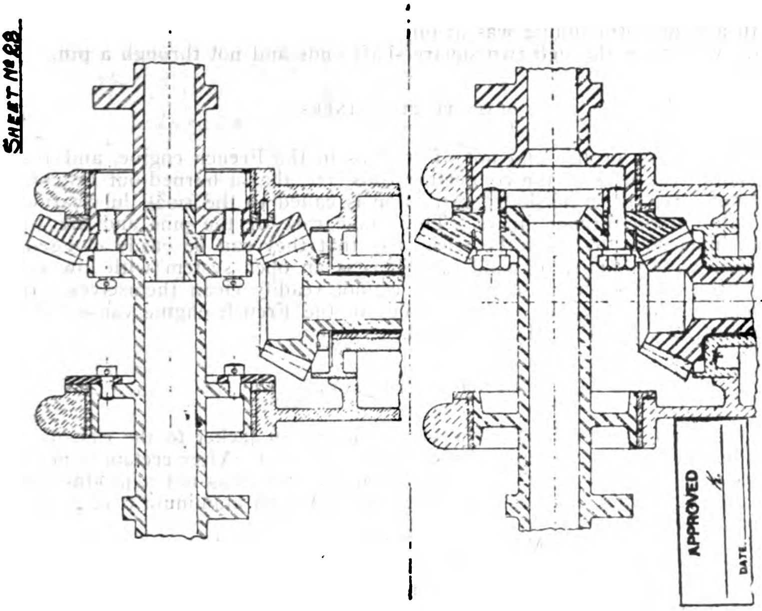


Fig 28

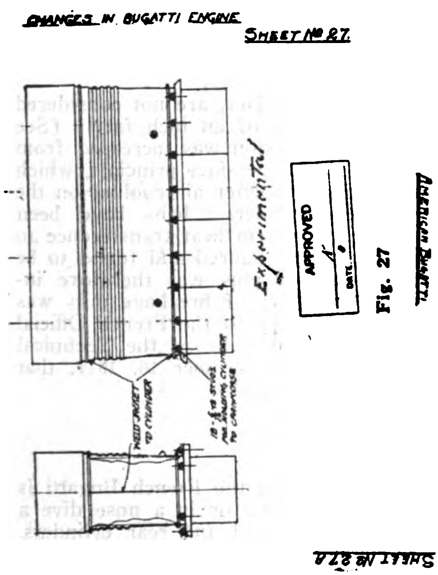


Fig. 27

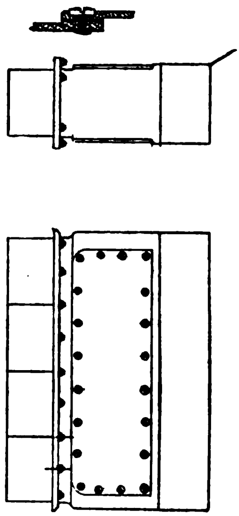


Fig. 27A

and then drilling the hole for pin N in vertical shaft. This was avoided in the American model by extending key to crank cheek ZZ fitting the gear directly to this same shaft. The key was accurately cut in relation to a tooth and uniform timing was at once obtained. The vertical shaft drive above was taken through two square shaft ends and not through a pin.

MAIN BEARING LINERS.

These are of bronze and not of steel as in the French engine, and for the same reason as in the connecting rods, viz, that a burned-out bearing will not seize the crankshaft. Attention is called to the small lubricating oil holes in the French liners. These take care of the pins and are of such dimensions .047 inch (see Fig. 26) that they can be easily clogged, whether with sediment or cold oil. This is an open system with low oil pressures, and hence these passages cannot readily clean themselves. It is possible that clogging of these holes in the French engine caused the wreck of this engine on the test stand.

OUTSIDE WATER JACKETS.

It was first contemplated to weld the sheet-iron jacket to the cast-iron cylinder by the oxyacetylene process. (See Fig. 27.) After cretain experiments were carried out, it was considered that this was not a production job and the arrangement as shown on Fig. 27A with aluminum covers will probably be adopted.

This examination by Mr. King and his engineers led them to the conclusion that the Bugatti engine as delivered by the French to the American government was not a commercial engine, as it was necessary to redesign the entire motor and to bring it in line with American practice.

PISTON AND PISTON PIN.

The five piston rings, namely, 3/32 of an inch face, are not considered commercial and were replaced by three rings ⅛ of an inch face. (See Fig. 31.) Further, the effective length of the piston was increased from 4 1/16 to 4⅞ and said piston was made of the slipper-face principle, which eliminated contact with the side walls and gives better air cooling on the hot unjacketed vertical sections between cylinders. Ribs have been eliminated and the pistol wall arranged for uniform heat transference to the jacket. The stresses on the piston ring were figured and found to be high, indicating a weakness. The diameter of pins was therefore increased from 63/64 to 1 1/16 inches. No report of breakage pins was received with engine, and upon receipt of the copy of the French Official Test received in May, 1918, from Lt. Col. Dunwoodie of the Technical Section, it was noted that in the test run of November 16, 1917, that piston pins were broken and three had to be replaced.

CARBURETORS.

The placing of the carburetor float chambers in the French Bugatti is not according to standard practice and in climbing or in a nose dive a different mixture is likely to result in the forward and rear cylinders. (See Figs. 33 and 34.) In the French engine one float chamber is forward of the jet and one is in the rear. In the American engine they are both brought to the rear. The American engine therefore gets a uniform mixture in all cylinders.

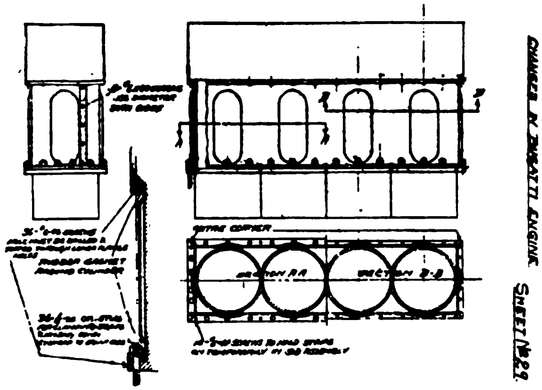

CHANGES IN BUGATTI ENGINE
SHEET No 29

Fig. 29

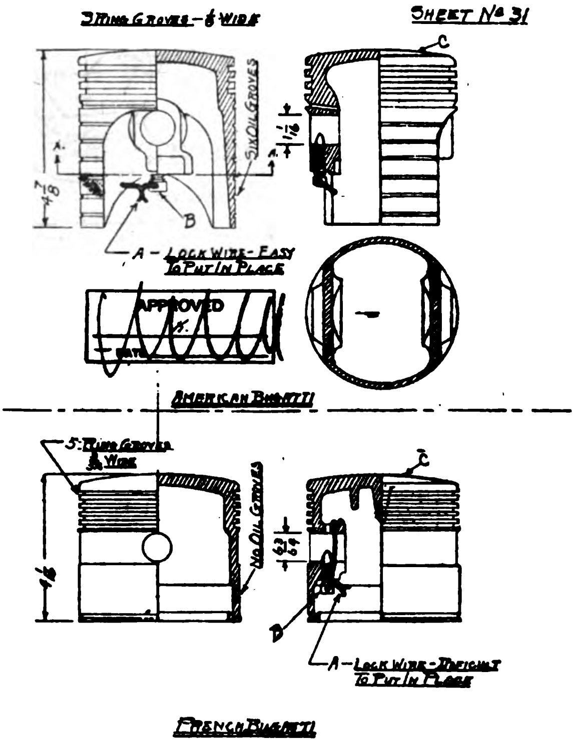

SHEET No 31
SIX OIL GROVES
B
A – LOCK WIRE – EASY TO PUT IN PLACE
APPROVED
AMERICAN BUGATTI
NO OIL GROVES
C
A – LOCK WIRE – DIFFICULT TO PUT IN PLACE
FRENCH BUGATTI

Fig. 31

IGNITION.

It was decided that two magnetos would be used. With the Splitdorf magneto a separate distributor is used mounted on the ends of the camshafts. The Simms magneto was used for half of the production, and these two magnetos were each equipped with their own distributors. A saving of over forty pounds was effected by this arrangement as against the use of four magnetos. Each engine is completely cross wired to each magneto; in other words, it is perfectly synchronized, and is such that

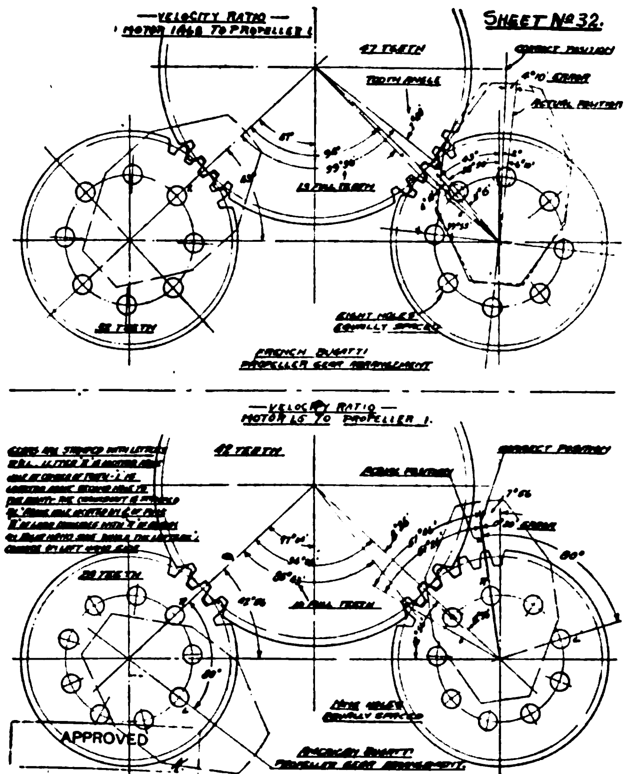


Fig. 32

each magneto taken separately will operate engine, and owing to this perfect synchronization the two magneto arrangement runs with less vibration than the four-magneto scheme. With the two magnetos and the distributor a starting magneto is supplied. This consists of a hand-operated independent magneto in the fuselage, which delivers a shower of sparks on the proper cylinder as selected by the distributor. Ease of starting is facilitated in this manner.

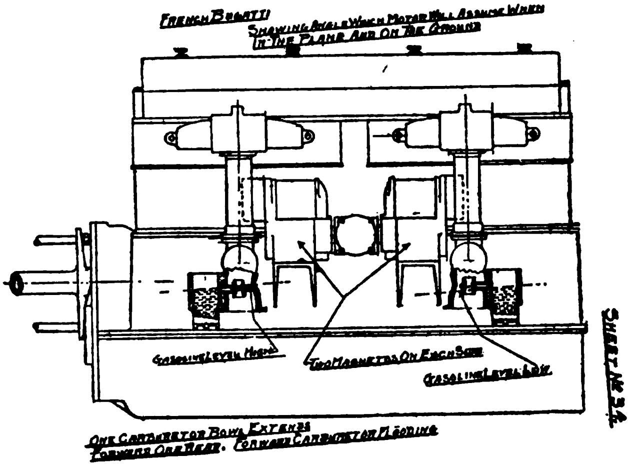

French Bugatti
Showing Angle Which Motor Will Assume When
In The Plane And On The Ground
Two Magnetos On Each Side
Gasoline Level Low
One Carburetor Bowl Extends
Forward One Rear.
Sheet No 34

Fig. 34

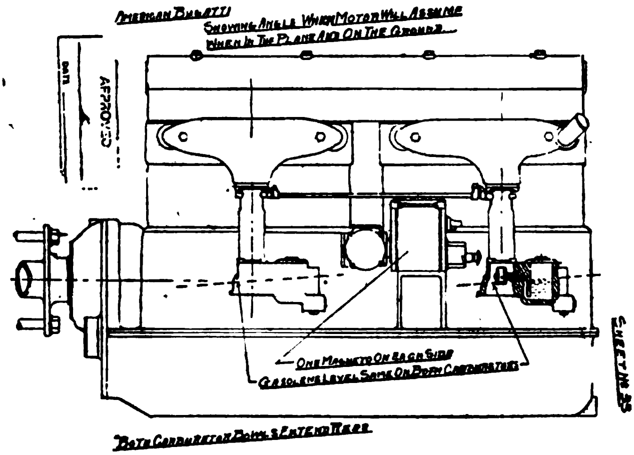

American Bugatti
Showing Angle Which Motor Will Assume
When In The Plane And On The Ground
Approved
One Magneto On Each Side
Gasolene Level Same On Both Carburetors
Both Carburetor Bowls Extend Rear
Sheet No 33

Fig. 33

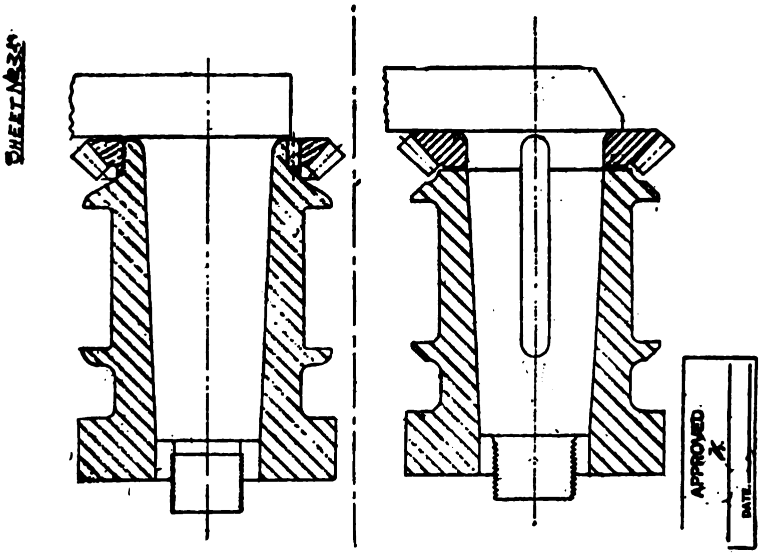

SHEET No 35
APPROVED
DATE.

Fig. 35

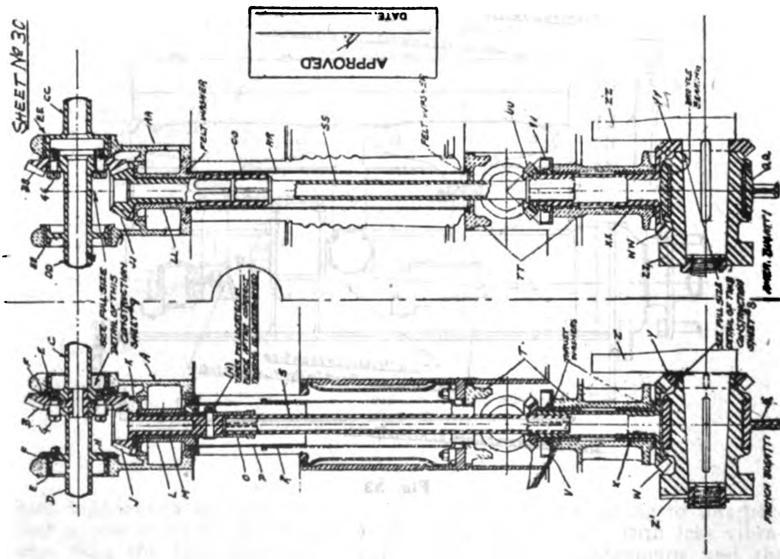

SHEET No 30
APPROVED
DATE.
FELT WASHER
THRUST WASHERS

Fig. 30

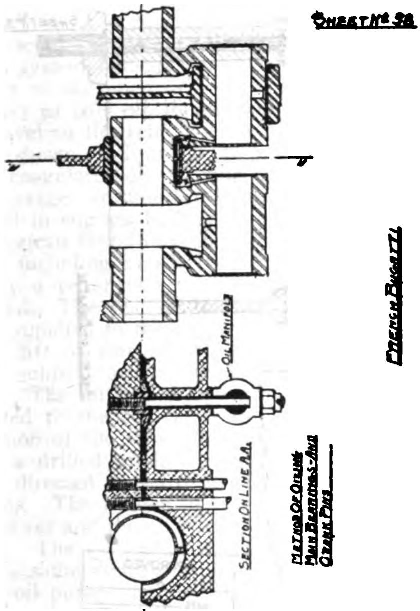

SHEET No 38
OIL MANIFOLD
SECTION ON LINE AA
METHOD OF OILING MAIN BEARINGS AND CRANK PINS
FRENCH BUGATTI

Fig. 38

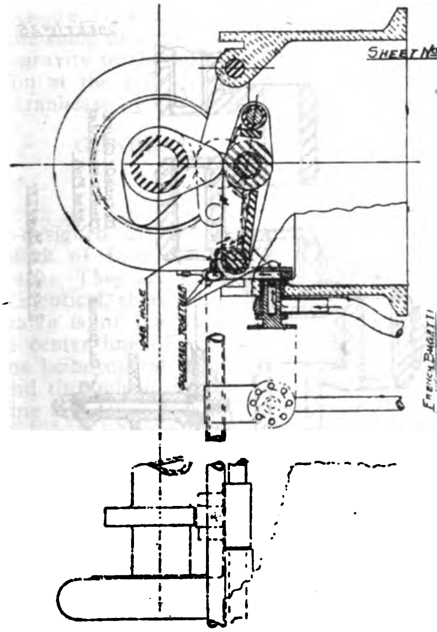

SHEET No 37
FRENCH BUGATTI

Fig. 37.

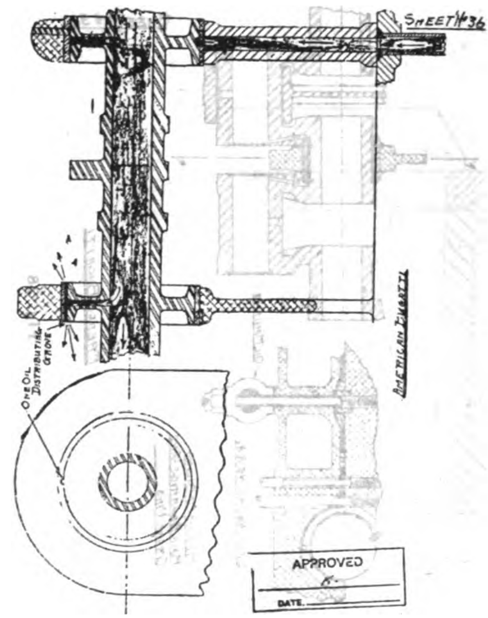

SHEET No 36
ONE OIL DISTRIBUTING GROOVE
AMERICAN BUGATTI
APPROVED
DATE

FIG. 40.

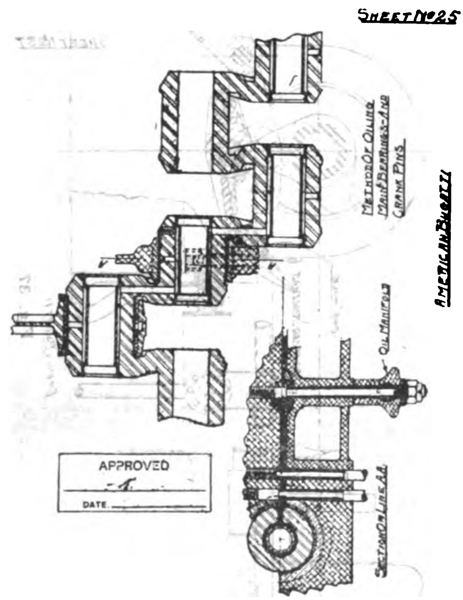

SHEET No 25
METHOD OF OILING MAIN BEARINGS AND CRANK PINS
AMERICAN BUGATTI
OIL MANIFOLD
SECTION ON LINE A.A
APPROVED
DATE

FIG. 39.

OILING SYSTEM.

The oiling system as applied to the French Bugatti engine can be considered an open-system non-pressure type, the pressure being determined by the diameters of the open orifices. (See Figs. 37 and 38.) In other words, with heavy or cold oil, the oil will take the path of least resistance and will not travel to the remote ends of oil leads. The small openings in bearings (as shown on Figs. 26, 37 and 38) become clogged with sediment, waste or coagulated oil. Owing to the pressure not being sufficient to clear these passages, trouble can be expected. Such trouble has already been experienced in engines built in this country. The fundamental principle in the American Bugatti engine was to obtain a true pressure system on all bearings, including camshafts. (See Figs. 39 and 40.) The oil to be controlled by a relief valve. This can be regulated and the proper pressure obtained. The seat of this valve is scored so that at all times a flow of oil is supplied to the gear faces and an additional flow will also come from the lift of this relief valve from its seat. All exterior oil pipes which are subject to breakage have been eliminated in the American Bugatti engine. The oil is fed upwards through drilled holes in crankcase and directed to the center of the camshaft through two openings. At each revolution of the camshaft a small groove in the bushing of this bearing passes a drilled oil hole lead to the center of camshaft. This squirt of oil is directed outward at each revolution over the cams, gears and rocker arms. The tie-bar in this engine is brought in contact with the front gear cover and the oil is led directly into same without the use of exterior piping. The oil led to the gage is further brought out at the rear end of this same member and an extra length of gage pipe is thus avoided. Two oil pumps are driven by the crankshafts and said pumps are mounted on the propeller gear housing. One pump is arranged to supply the pressure system, the duplicate pump is used to clear the forward end of the sump in a nose dive. In climbing, the oil is led out of the crankcase by gravity to the oil supply tank. Pipe into said tank being led into a position at the rear of the tank in order that the oil cannot flow back in the crankcase by gravity.

GENERAL DESCRIPTION.

CARBURETION.

Four specially-designed Miller carburetors are used (Fig. 41), each supplying one block of four cylinders through separate water-jacketed manifolds (Fig. 42). They are set low so that gravity feed may be used and all four are identical, there being no rights or lefts.

The throttle valve is of the barrel type, the axis of all valves being parallel with the center line of the engine, the two carburetors on each side of the engine being operated by one shaft which is connected to the valves at each end through adjustable couplings. The shafts on the two sides of the engine are connected so that all four valves move in unison, the valve opening being synchronized by means of the adjustable couplings.

Gasoline from the tank enters the carburetor through the elbow (Fig. 41), passing through the strainer, thence into the float chamber, flowing out through the four 5/32 inch holes in the lower end of the needle-valve seat.

When the gasoline reaches the proper level the rising of the cylindrical brass float lowers the needle valve onto its seat, thus stopping the flow. From the float chamber the gasoline enters the lower 3/16 inch hole in jet holder. There are seven of these jets, drill sizes being as follows: No. 76, which is the idling jet, No. 76, No. 75, No. 71, No. 68, No. 57, No. 53. These jets progressively come into action as the throttle is opened.

Gasoline is drawn into the jet through the small hole in the bottom of the threaded end, mixing with a certain amount of air sucked in through the four holes drilled in the barrel of the jet just above the threaded portion. This air is taken from the outside through the upper 3/16 inch hole in jet holder and passes down around the outside of the jet to the four holes mentioned above. The major portion of the air enters the carburetor through the lower end of the venturi, which is 3 inches in diameter, passes up around the jet bar holder, combining above this with the rich mixture from the jets to form the proper mixture for combustion.

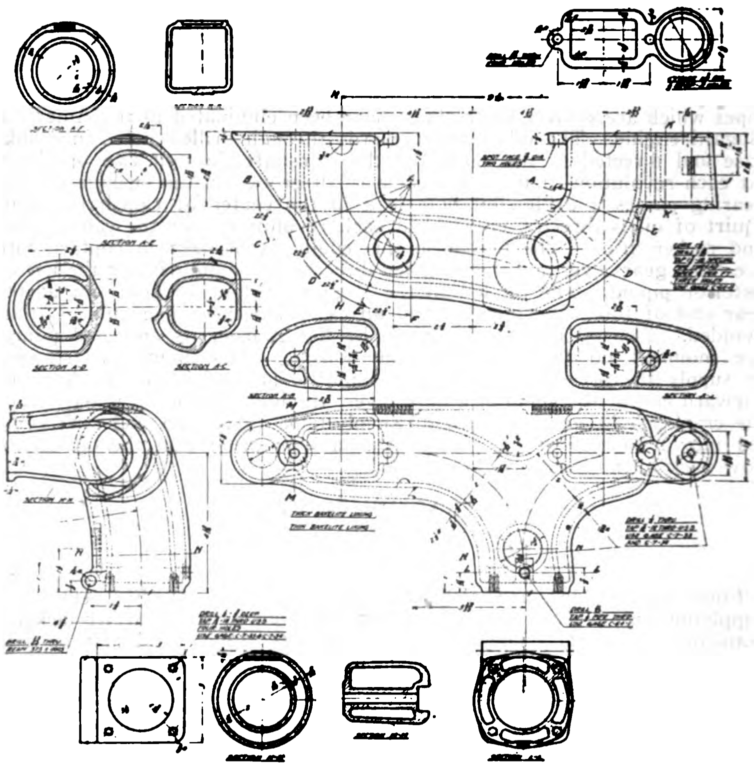


Fig. 42—Details of Gas Inlet Manifold—Front right and rear left

Assembly of the altitude valve is shown on drawing (Fig. 43). This valve operates by turning the lever which is attached to the altitude control valve. This valve has two openings in its seat which when in the open position register with two similar openings in the stationary cover, thus making two free passages to the outer air, the size of this passage being governed by the position of the lever.

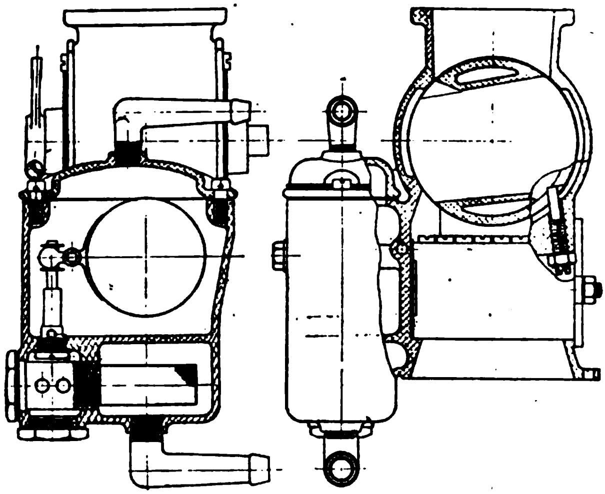


Figure 41—Carburetor Assembly

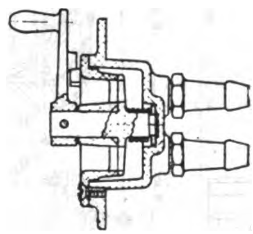


Figure 43 — Altitude Control Valve Assembly

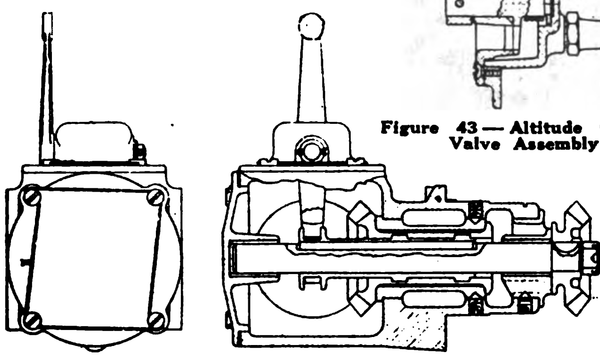


Figure 45—Magneto Gear Housing Assembly

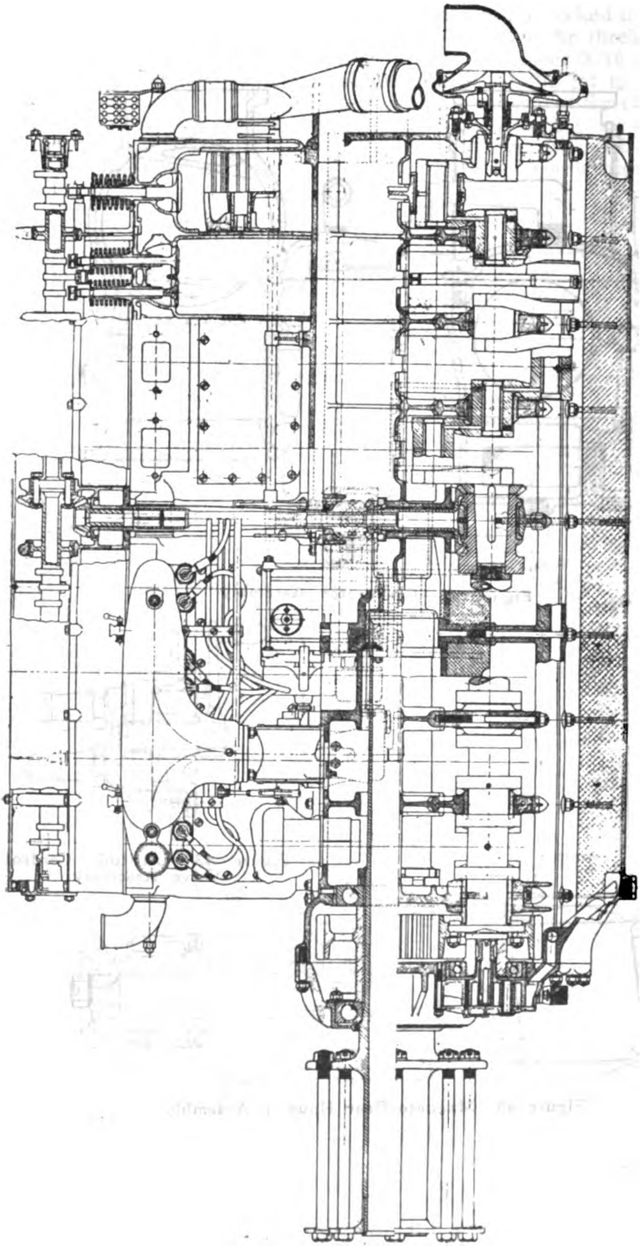


Fig. 44—The 410 H.P. King-Bugatti engine—longitudinal section

There are four outlets (Fig. 43), one of which connects to each of the four elbows, opening directly into the top of the float chamber. The float chamber is always in direct connection with the venturi through a 5/64 inch drilled hole opening into the venturi about ¼ inch above the jet holder and into the float chamber well above the gasoline level. Opening the altitude control valve decreases the vacuum in the float chamber, thus increasing the flow of gasoline through the jets.

IGNITION SYSTEM.

Ignition is by four Dixie 800 magnetos, two on each side of the engine, driven from the vertical camshaft driving shaft by bevel gearing as shown on drawing Fig. 44. All magnetos turn clockwise.

Two Titan A C spark plugs are used per cylinder located in the side of the combustion chamber.

The rear magneto on the right-hand side supplies current to the rear plug in each of the eight left-hand cylinders, the front magneto on the right-hand side supplying current to front plug in each of the eight-right-hand cylinders.

The same arrangement is followed with the magnetos on the left-hand side so that the two magnetos on either side will fire all sixteen cylinders.

Magnetos are set for a maximum advance of 38 degrees.

The bevel gear on the magneto shaft, Fig. 44, is fitted on a taper with a key. The gear has eight keyways, six spaced 48 degrees, one spaced 42 degrees and one spaced 30 degrees. This in combination with the gear teeth allows the magneto to be set within 1½ degrees on the crankshaft.

The magneto advance and retard mechanism is shown in Fig. 45. Gear inside meshes with the gears on the magneto shafts. This gear has four internal spiral grooves sliding over splines on the sleeve which is keyed to the driving shaft, but may be moved along the shaft by lever. The movement of this sleeve revolves the magneto-driving gear in relation to the shaft-driving gear, thus advancing or retarding the magnetos. The levers on the two sides of the engine are operated from one shaft located above the crankcase between the cylinder blocks, the connections to the levers being through adjustable yokes so that the magnetos may be synchronized.

OILING SYSTEM.

Oiling is by means of pressure feed and spray. There is one pressure and one scavenging pump both of the rotary gear type. These are located at the front of the engine, driven directly from the crankshafts through a pin and slotted coupling, Figs. 46 and 47. This coupling is squared to the pump shaft, but is not pinned, thus relieving the shaft of any end driving pressure. The gears in both pumps are the same except that the scavenging pump gears have a wider face.

Oil, after passing through a strainer, is drawn from the supply tank by the pressure pump which is driven from the right-hand crankshaft. This oil is forced into the pressure line running the entire length of the crankcase as indicated in Fig. 48. An adjustable pressure-regulating valve is located in the crankcase front gear cover. It is of the poppet-valve spring-seated type and discharges the excess oil directly onto the propeller shaft driving gears. This valve is generally set so that the pressure gage which is connected to the rear end of the main oil line in the crankcase registers about 30 pounds. This valve has holes drilled through the head so that there is always a certain amount of oil discharged onto the gears.

From the pressure line the oil passes up around the studs which hold this line in position to an oil passage cut along the top surface of the crankshaft bearing cap, see Fig. 48. For the center crankshaft bearing

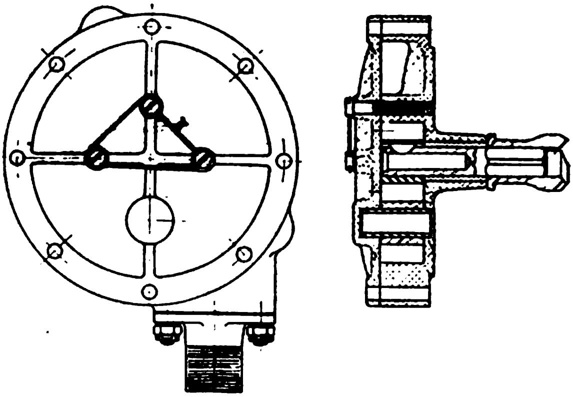


Figure 46—Oil Pressure Pump Assembly

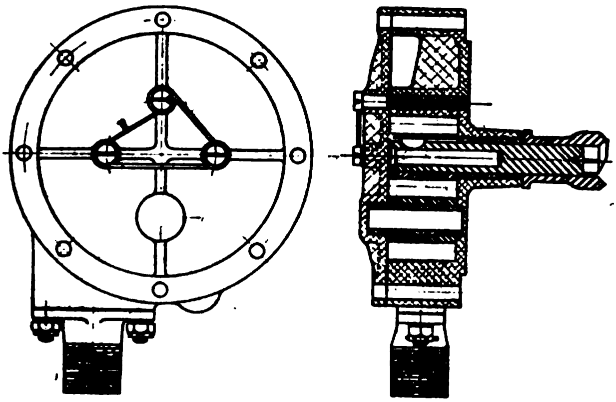


Figure 47—Oil Suction Pump Assembly

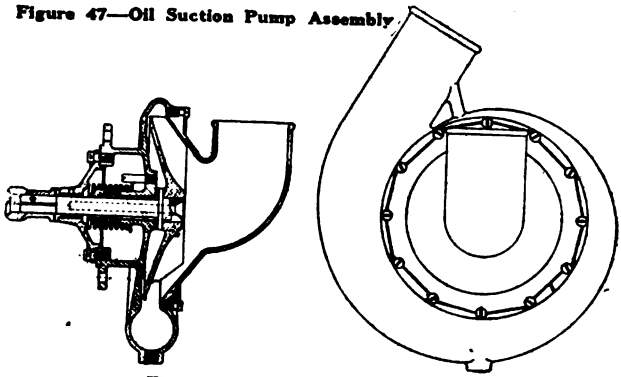


Figure 48—Water Pump Assembly

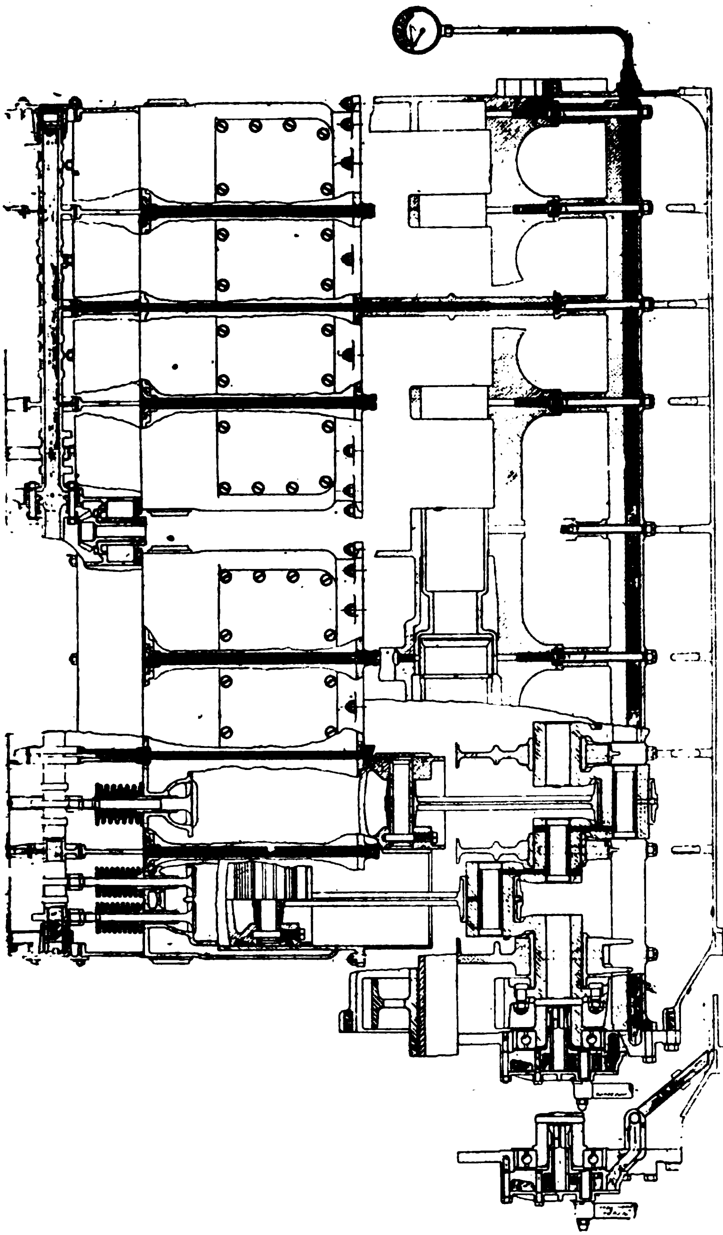


Fig. 48—Longitudinal section of the King-Bugatti engine, showing the oiling system

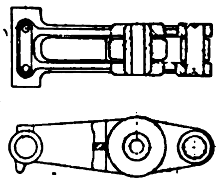
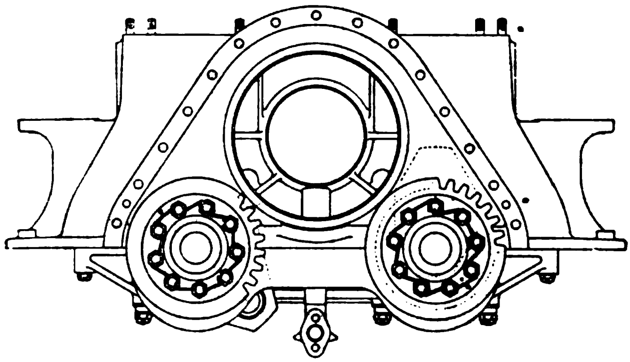
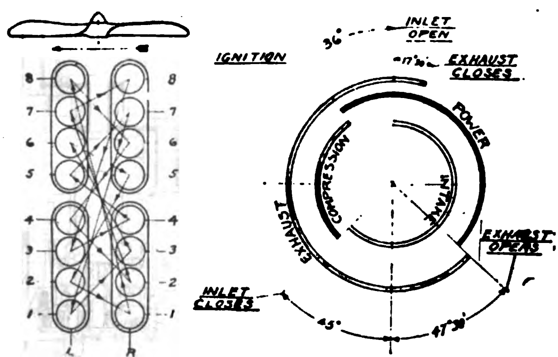


Figure 50—Valve Rocker Assembly



Crankshaft Assembly Chart



Timing Chart

this oil is carried through a drilled hole as indicated in Fig. 51. These passages carry the oil to all the main crankshaft bearings. All the main crankshaft bearings and pins are hollow. Main crankshaft bearings No. 2, No. 4, No. 6 and No. 8 have a 3/16 inch radial hole drilled entirely through them. All the crank-pin bearings have a 3/16 inch radial hole drilled from the inside to the central hole. A 13/64 inch hole is drilled in the web both sides of main crankshaft bearings No. 2, Nos. 4, 6 and 8 connecting the central hole in the main and pin bearings. A copper shell is placed in these central holes and the ends spun over, making an oiltight joint. These shells are necked in the central portion so that a tubular oil space is left as indicated in Fig. 48.

Oil from the passages in the crankshaft bearing cap is forced into this tubular oil space through the 3/16 inch holes which register with this passage twice per revolution. From here the oil passes to the pin bearings, the leakage from these bearings being thrown on the cylinder walls and gudgeon pins, thoroughly lubricating these parts.

Four vertical holes are drilled in the crankcase web connecting with the oil passages in the crankshaft bearing cap No. 3 and No. 7. These holes register at the top of the crankcase with copper tubes which pass through the cylinder water-jacket space, registering at the top of the cylinder block with four holes drilled in the webs of the crankshaft housing. These holes register with an oil groove of 3/64 inch radius cut entirely around camshaft bearings No. 3 and No. 8, right and left hand. A No. 35 drill hole connects the oil grooves with the interior of the hollow camshaft. Oil is thus carried under pressure to the hollow camshaft. From the hollow camshaft, the oil passes to camshaft bearings No. 1, No. 2, No. 4, No. 5, No. 6, No. 7, No. 9 and No. 10 through a No. 35 drill hole. Camshaft bearing bushings No. 2, No. 4, No. 7 and No. 9 have a 3/32 inch 1/32 inch oil groove cut full length of the bearing surface, the drilled hole in the camshaft bearing registering with this groove once per revolution causing a small stream of oil to shoot out both sides of the bearing thoroughly lubricating the cams, valve rocher shaft, rollers and valve stems. Camshaft bearing bushings No. 5 and No. 6 have a 3/32 inch x 1/32 inch oil groove cut from ¼ inch of the outer edge to the inner edge, the drilled hole in the camshaft bearing registering with this groove once per revolution. Oil from this groove in the rear bearing lubricates the thrust surface of the camshaft bevel gear while the small stream from the front bearing thoroughly lubricates the camshaft and camshaft driving gears and the camshaft driving gear bearings.

A 3/16 inch hole is drilled in the crankcase web connecting with the oil groove in No. 6 crankshaft bearing cap and registering with a 1/16-inch drilled hole in the propeller-shaft rear bearing bushing. This thoroughly lubricates this bearing. The oil flowing from this bearing returns to the sump by gravity.

The camshaft and magneto driving gears in the crankcase are lubricated by spray. The gearing in the camshaft housing is packed in grease.

The crankshaft and propeller-shaft ball bearings are lubricated by spray.

Oil which drains to the bottom of the camshaft housing is returned by gravity to the crankcase through twelve pipes passing through the cylinder water-jacket space.

Oil which drains to the front end of the sump is returned to the oil tank by the scavenging pump.

Oil which drains to the rear end of the sump is returned to the oil tank by gravity.

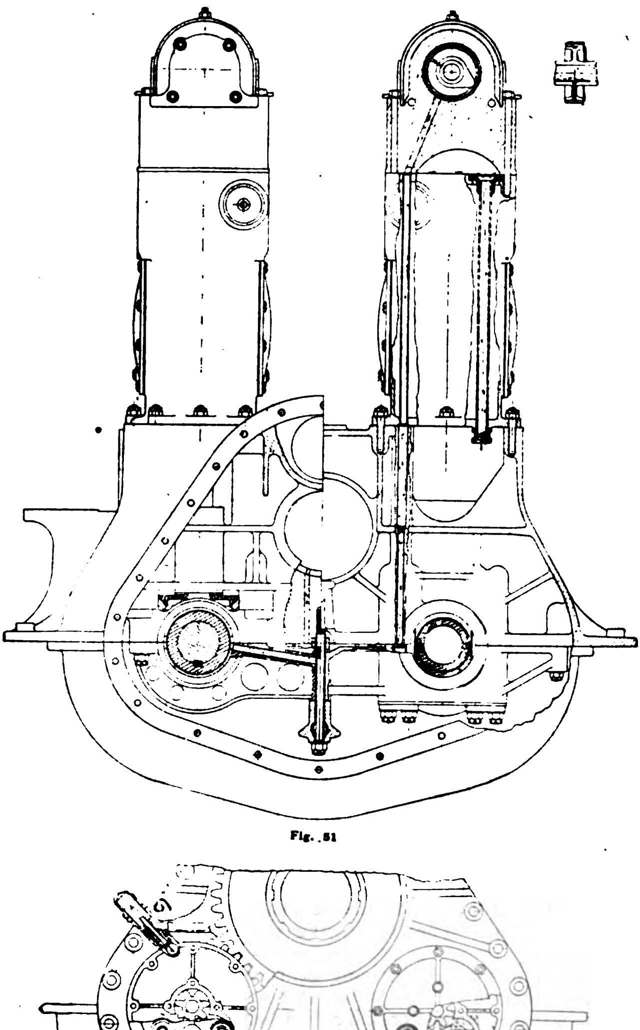
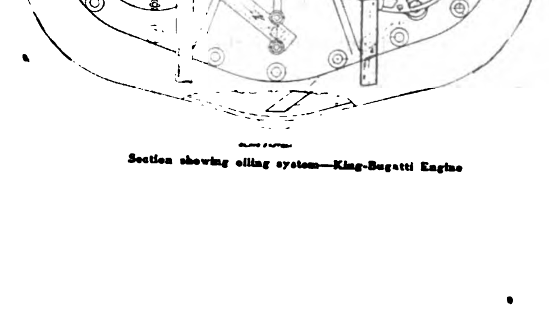


Fig. 51



Section showing oiling system—King-Bugatti Engine

COOLING SYSTEM.

Cooling water is circulated through the engine by means of a centrifugal pump, see Fig. 49, driven from the rear end of the left-hand crankshaft by a pin and coupling the same as used on the oil pump.

The cooling system from the pump inlet to the outlet elbows on the front cylinders holds four and one-quarter gallons of water.

The pump impeller is 5⅞ inches diameter with eight vanes, the web being drilled with eight ⅜-inch holes on a circle of 2 inches diameter to equalize the water pressure.

The pump shaft is packed with a graphited asbestos rope packing, automatically held under compression by a coiled spring acting on the gland.

The pump shaft is hollow, the rear end being in direct communication with the water in the pump case. Water entering the shaft is forced out to the shaft rear bearing surface through a ⅛-inch hole. Any leakage of water past the asbestos packing is drained outside of the crankcase through a ⅝-inch cored hole in the water-pump body. The front bearing on the pump shaft is lubricated by spray from the crankcase which collects on the shaft bushing support and drains down into a ⅛-inch hole leading to the bearing. Any oil leakage from the front end of this bearing returns to the sump, any slight leakage from the rear end of the bearing is drained outside of the crankcase with the water leakage from the rear bearing.

There is one water inlet to the pump 2¼ inches inside diameter while the single outlet is 2 3/16 inches diameter. Water from the pump is forced up into an aluminum pipe with one branch leading to the rear end of each of the rear cylinder blocks, water entering the cylinders at the top of the water jacket on the exhaust side. A certain amount of the water circulates through the inlet manifold jacket, the remaining filling the cylinder water jacket space.

Cylinder blocks are cast with integral water jackets except the sides below the inlet and exhaust ports which are covered by an aluminum water jacket plate held in position by screws.

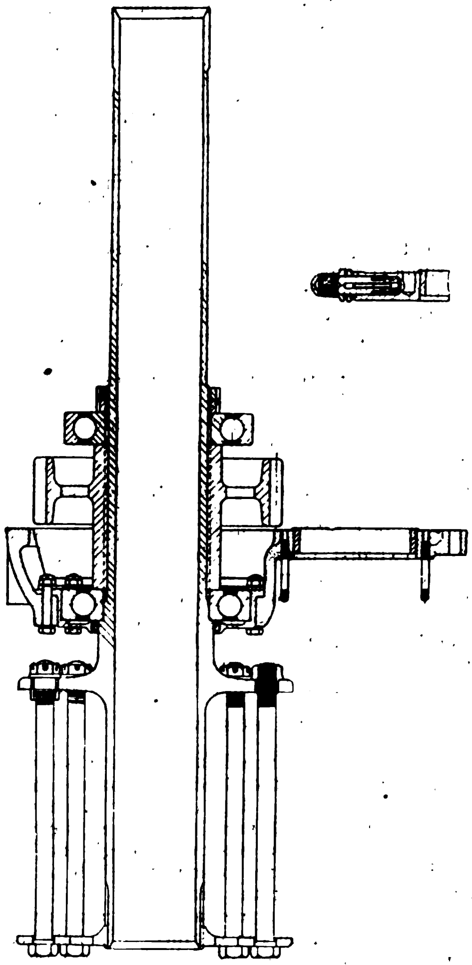
The construction of the water passages in the head of the cylinder is such that the valve stems and exhaust passages are very thoroughly cooled.

All four cylinder blocks are identical, the water passing from the rear to the front cylinder blocks through openings located similar to the inlet opening, leaving the front blocks through another similar opening.

PROPELLER SHAFT.

The propeller shaft is driven through a spur gear splined to the shaft meshing with a gear on the front end of each of the crankshafts, both the crankshafts turning clockwise. The propeller shaft is hollow. Provision is made for mounting a 37 m.m. cannon at the rear end of the crankcase, the barrel of the cannon passing through the hollow propeller shaft. This shaft is carried in three bearings, a ball bearing either side of the gear and a plain bearing at the rear end.

The front gear cover, the ball bearings and the gear are assembled complete as a unit before mounting in the engine. The ball bearings are No. 6219 Monarch Special, Width Hess Bright, being narrower than the standard bearing. The front bearing is mounted in the gear cover and takes all the propeller thrust as well as a certain part of the radial load. The rear bearing slides into a retainer in the crankcase, being free to move endwise, carrying radial load only. The hub of the gear acts as a spacer for the ball bearings, they being held in position on the shaft by two nuts with a locking plate between. Mounting is such that the ball bearings and gear are easily and positively assembled there being no danger of injuring the ball bearings by screwing the retaining nuts too tight.



Propeller shaft and front cover assembly

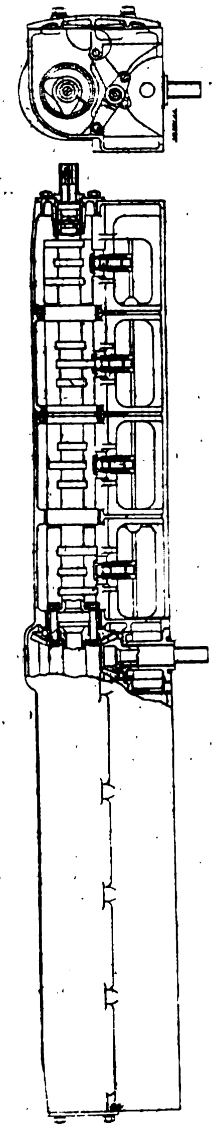
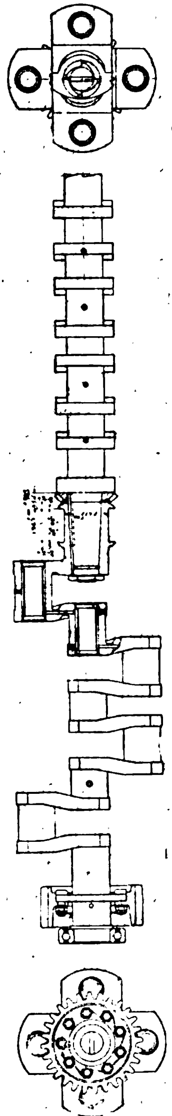


Fig. 82—Camshaft Housing Assembly—right



Crankshaft Assembly—right hand

CAMSHAFT DRIVE.

There are two overhead camshafts, each one running full length of the two cylinder blocks. Each shaft is made of two separate shafts joined near the center by six bolts passing through flanges on the shafts. These bolts also hold the camshaft gear in position. Each complete shaft is carried in ten plain bearings in a removable camshaft housing. The two housings are of aluminum and each one is bolted directly to the top of the cylinder blocks without the use of a gasket by twenty ¼-inch studs.

The first camshaft housings had a sheet aluminum cover as indicated on Fig. 51, a later housing is made entirely of cast aluminum with a cast aluminum cover plate at the side running the entire length of the housing. See Fig. 52.

Each camshaft is operated through a bevel gear driven by a vertcial shaft between the two cylinder blocks which in turn is driven by a bevel gear on the crankshaft. The gear on the crankshaft is pressed in position and in addition is held by a key. The keyway is cut in a definite position with relation to a marked tooth. The thrust of this gear is taken by the crankshaft center bearing. This gear meshes with a bevel gear which drives the vertical shaft; the thrust of the gear being taken on a bronze Babbitt lined bushing pressed into the aluminum crankcase. This gear has a long shank which acts as a bearing for the gear and has a fine external thread cut at its upper end. This upper end has a square broached hole, the hole having a definite relation to a marked tooth of the gear. The bearing for the upper end of the shank of this gear is a bronze bushing Babbitt lined pressed into the crankcase. The upper end of the bearing is cupped to form an oil well which catches the spray in the crankcase which is led to the bearing. The magneto driving shaft driving gear, see Fig. 44, is screwed into the thread at the upper end of the shank, the adjustment of the gear being obtained by means of the thread. This gear has a square broached hole the same size as the hole in the shank of the gear it is screwed to and is locked in position by dropping the camshaft driving shaft into position, this shaft having a squared section at both ends, the lower end fitting into the square hole in the magneto driving shaft driving gear and the square hole of the shank of the camshaft driving shaft gear. The gear on the camshaft meshes with a gear having a long shank with a squared section at its lower end, the square being cut with a definite relation to a marked tooth on the rear. This gear takes a bearing in an aluminum Babbitt-lined bushing which is held in position in the camshaft housing by a large flat-head screw. This bushing also takes the thrust of the bevel gear. The upper end of the vertical camshaft driving shaft and the lower end of the camshaft driving gear are connected by a coupling consisting of a square piece of steel with a square broached hole fitting over these square ends.

The teeth of all the gears being marked for the meshing position they may be easily assembled in the proper positions for correct timing.

VALVE MECHANISM.

The valves are operated by an overhead camshaft through a rocker arm. These rocker arms, Fig. 50, are pivoted on steel shafts which are slid into drilled holes along both sides of the camshaft housing as indicated in Fig. 50. There are four of these rods to each housing, each one-half the length of the housing. The ends of the rods butt together at the center, the outer ends being flush with the ends of the camshaft housing and covered by the end bearing support for the camshaft when this is bolted in position. The rods are thus prevented from moving

lengthwise. They are of a tight enough fit in the drilled hole in the camshaft housing so that there is no turning motion. The outer end of each rod is tapped for a wrench to be used in withdrawing the rods from the housing.

Each rocker arm operates one valve, they are forgings and the pivot bearing of the arm on the rod in the camshaft housing is not bushed. The cam operates on the large roller, Fig. 50, which is of hardened steel taking a bearing on a hardened-steel pin. This pin is held in the rocker arm by spinning the metal of the rocker arm around the beveled end of the pin as clearly shown in Fig. 50. The small roller operates directly on a cap placed over the end of the valve stem. This roller is of hardened steel and takes a bearing on a hardened-steel pin the ends of which are soft and spun over into a bevel at the outer edges of the hole in the rocker arm.

The end of the valve stem has a cap slipped over it as is indicated in Fig. 44, the upper end of the cap being hardened. The proper clearance .015 inch for both the inlet and exhaust valve between the end of this cap and the roller in the rocker arm is obtained by placing shims in the cap. Three different steel shims are used of .003 inch, .005 inch and .010 inch thickness, the first being octagonal, the second hexagonal and the third round, so that the different thicknesses may easily be picked out by eye.

The upper valve-spring retainers have a central tapered hole, the large end of the hole being on top. The vale stems are necked and a tapered split collar is slipped into the necked portion of the stem, large end up. This taper is the same as the hole in the spring retainer. The pressure of the valve springs forces the retainer against the tapered collar which is prevented from moving by the shoulder on the valve stem thus locking the retainers into position.

VALVES.

The inlet and exhaust valves work in cast-iron guides pressed into the cylinders. Liberal water space is provided in the cylinder head in the neighborhood of these guides so that the valve stems are well cooled.

The exhaust valve stem is hollow from the head to within a short distance of the necked portion at the upper end. The hole is closed at the head end by a short threaded plug screwed in below the surface of the valve, the recess then being filled level with the surface of the valve by welding. This closes the hole tightly and locks the plug in position. The lower end of the exhaust valve stem is of larger diameter than the upper end. Both the large and small diameters take a bearing in the valve guide as indicated on Fig. 44. At the shoulder formed by the junction of the two sizes of stem three 3/32 inch holes are drilled at an angle of 30 degrees with the axis of the stem sloping towards the head of the valve and connecting with the drilled hole in the stem. At the upper end of the stem just below the necked portion a 5/32 inch hole is drilled through the wall of the stem. The movement of the valve up and down in the guide causes a pumping action, the transfer of air within the valve stem being thought to cool the stem to a certain extent. This drilling also lightens the valve.

PISTON.

The piston is of aluminum and has two ring grooves above the gudgeon pin. Two 1/8-inch wide rings are placed in both of these grooves. The rings require a pressure of ten to twelve pounds applied on a diameter

at right angles to the slot to bring the ends of the 30-degree slot to within .010 inch of the closed position. The lower ring is beveled, being placed in the groove with the sharp edge down so that it acts as a wipe ring forcing the oil on the down stroke into an oil groove cut just below the ring groove. The land between the bottom of the ring groove and the oil groove is 1/64 inch wide and it is 1/32 inch smaller diameter than the land above this ring groove. This forms a free passage for the oil to the oil collecting groove. Eight 3/32-inch holes are drilled around the piston connecting the oil groove with the interior of the piston. These holes slope down at an angle of 60 degrees with the axis of the piston carrying the oil wiped from the cylinder wall to the interior of the piston. A slot ⅛ inch wide x 1/16 inch deep is cut from the oil groove to the gudgeon pin hole at both ends thus lubricating these bearings. Two ½-inch wide x 1/32-inch deep oil distributing grooves are cut around the piston, one about 3/16 inch above and one about 5/16 inch below the center line of the gudgeon pin.

GUDGEON PIN.

The gudgeon pin floats in both the connecting-rod bushing and the piston. Bearing in the piston is directly on the aluminum. It is held from endwise motion by an aluminum plug pressed into each end. See Fig. 44. These plugs are drilled axially with a ⅜-inch hole, which allows a certain amount of oil from the cylinder wall to enter the hollow gudgeon pin. The ends of these plugs where they bear against the cylinder wall are turned to a spherical seat of a radius equal to the radium of the cylinder bore. In operation the gudeon pins turn more or less. Two ⅛-inch holes, in line, one ¾ inch each side of the center of the pin, are drilled through the wall of the pin. These holes when in a certain position register with an oil groove in the connecting-rod bushing allowing a certain amount of oil to enter the hollowed gudgeon pin and when in the lower position allow the oil which is pocketed in the pin to run out onto the connecting-rod bearing.

CONNECTING ROD.

The connecting rod is a steel forging machined all over. It is of eye cross-section.

The small end bearing is a bronze bushing pressed in position, having one straight oil groove, 3/32 inch x 1/32 inch deep running to with ¼ inch of the ends of the bushing. This groove is in communication with a 13/32-inch drilled hole in the connecting rod through a ⅛-inch hole. Oil collects in the pocket formed by the 13/32-inch hole, and is led to the bearing through the ⅛-inch hole.

The big end of the connecting rod is fitted with a bronze bushing which is babbitt line. This end is split at right angles to the rod on the center line of the bearing, the cap being held in position by two ⅜-inch chrome nickel-steel bolts. The bushing is relieved at the line by 5 grooves, 3/16 inch wide, about ¾ inch long, 3/64 inch deep, at the parting line cut on a 7/16-inch radius with a center on the parting line. The rod half of the bushing has no oil groove, the cap half has a circular oil groove entirely around it cut on the center line of the bushing and registering with the oil-feed hole in the crank pin. This groove is ⅛ inch wide x 1/32 inch deep. The bushing in the cap is kept from turning by a dowel pin with an enlarged head. This head enters a countersunk hole in the cap, the small end entering the hole in the bushing when it is placed into position. This end is of such a length that it does not project through the bushing. The pin is thus locked in position and cannot drop out or rub on the shaft.

CRANKSHAFT.

The crankshaft is made in two pieces connected at the center by a taper and key drawn up with a nut. Each section of the shaft forms a four-cylinder shaft with the throws all in one plane, the throws of the two sections being assembled at right angles. In assembling, the rear end of the front half is immersed in boiling water, the tapering end of the rear section which is cold is then slipped into position and the parts drawn together by the nut using a long handled wrench.

The rear end of each complete shaft has clutch teeth cut on it for attaching a starter.

All end thrust coming on the shaft is taken by the center or No. 5 bearing all the other bearings having about 1/16-inch clearance at both ends.

All bearings, including the connecting rod bearings, with the exception of the center main bearing are undercut. This results in a total shortening of the shaft of approximately 4 31/32 inches. This of course results in a considerable saving in the weight of various parts while still allowing ample bearing surface.

The crankshaft main bearings are bronze bushing, babbitt lined. These bearings are not relieved at the parting line and there are no oil grooves excepting in the lower half of bearings No. 1, No. 2, No. 3, No. 4, No. 6, No. 7, No. 8. These have a 3/32-inch wide x 1/32-inch deep circular oil groove entirely around them. This groove registers with the oil into the hollow crankshaft bearing. As this oil hole is drilled through both walls of the bearing on a diameter and as the oil groove in the bushing extends through 180 degrees there is always a free passage for the oil from the bearing into the hollow crankshaft.

In assembling the completed crankshafts in the crankcase, they are placed in such a relation to each other that if No. 8 throw left is on top dead center No. 8 throw right will be 45 degrees past bottom dead center. Both cranks turn clockwise viewed from the rear of the engine.

The propeller driving gears are bolted to the crankshaft with nine bolts equally spaced, the bolt holes being drilled in a certain relation to the gear teeth. This makes it possible to use the same gear on either shaft with a maximum error in the setting of the shafts of 20 minutes. The flange on the crankshaft and the gears are marked as indicated in Fig. 51 for the proper position of assembly of the gears on the shafts.

CYLINDERS.

The cylinders are of iron cast in blocks of four. They are bolted directly to the top of the crankcase without the use of a gasket. The water jacket is cast integral with the exception of the sides of the cylinder block below the inlet and exhaust ports, which are covered with a cast aluminum plate attached with screws. A gasket is used between these plates and the cylinder.

Cylinders are cast with separate exhaust ports. One inlet port supplies two cylinders. Two inlet and one exhaust valves are used.

The entire combustion space is machined with the exception of a very small recess near the inlet valve seat.

The tops of the cylinder blocks are machined, making an oiltight joint with the camshaft housing without the use of a gasket.

Provision is made for a liberal circulation of water in the neighborhood of the valve ports, seats and guides overcoming valve trouble.

Spark plugs, of which there are two per cylinder, are located at the side of the combustion chamber in close proximity to the inlet valves and are well cooled by the circulating water.

CRANKCASE.

The crankcase and oil pan are of cast aluminum. The case is well ribbed. All bearings are supported from the upper part of the case, the bearing bushings being held in place by caps which extend almost the full width of the case. Each cap supports two bearings. They are all of cast aluminum with the exception of the center bearing cap, which takes all the thrust of the crankshaft. This is a steel forging.

The oil pan has a web running its entire length along the center line of the engine. This greatly assists in preventing the current of air caused by the revolving crankshafts from drawing up the oil which is constantly draining into the pan. There is also a cross web between each cylinder block which runs well up the sides of the oil pan. There cross webs have an opening at the bottom at their center, allowing the oil to drain to the ends of the pan on either side of the web running lengthwise.

A breather is attached to the top of the crankcase near the front end between the cylinder blocks.

Provision is made at the rear end of the case for attaching a 37 m.m. cannon, the barrel of the cannon projecting through the hollow propeller shaft.

GUN CONTROL.

Provision is made for attaching two gun-control mechanisms, one at the rear end of each of the camshaft housings driven directly from the camshafts through a slotted coupling. When this control mechanism is used it is inserted in place of the camshaft housing end bearing cover.

TACHOMETER DRIVE.

The tachometer drive may be taken from either end of the two camshafts. It operates at camshaft speed through a slotted coupling as indicated in the illustration.

When the gun control mechanism is used the tachometer drive is taken from the rear end through the same type of coupling as when driven directly from the end of the camshaft.

GENERAL DATA

Number and arrangement of cylinders,	16 vertical, 2 rows of 8 in blocks of 4
Material	Cast iron.
Bore	4.33" 110 m.m.
Stroke	6.3" 160 m.m.
Stroke-bore ratio	1.455 : 1
Area of one piston	14.725 sq. in.
Total piston area	235.6 cu. in.
Swept volume of one cylinder	92.768 cu. in.
Displacement of motor	1484.288 cu. in.
Compression ratio	5 : 1
Normal brake H.P.	410 at 2,000 r.p.m.
Type of valve gear	Overhead camshaft and valve rockers.
Number of carburetors	Four Miller special.
Ratio propeller to crankshaft speed	666 : 1

VALVES.

Number per cylinder	Two inlet and one exhaust.
Outside diameter, inlet	1.535

Outside diameter, exhaust	2.263
Port diameter, inlet	1 27/64"
Port diameter, exhaust	2 3/64"
Width of seat, inlet	.057"
Width of seat, exhaust	.108"
Angle of seat	10"
Valve lift, inlet	.653"
Valve lift, exhaust	.700"
Diameter of stem, inlet	.357"
Diameter of stem, exhaust (large)	.591"
Diameter of stem, exhaust (small)	.4355"
Length of valve, inlet	5 17/64"
Length of valve, exhaust	5 13/32"
Number of springs per valve, inlet and exhaust	2 concentric.
Length of spring in position, inlet (small)	2 7/64"
Length of spring in position, inlet (large)	2 15/64"
Length of spring in position, exhaust (small)	2 27/64"
Length of spring in position, exhaust (large)	2 27/64"
Mean diameter of coils, inlet spring (small)	.57/64"
Mean diameter of coils, inlet spring (large)	1 9/32"
Mean diameter of coils, exhaust spring (small)	1 9/32"
Mean diameter of coils, exhaust spring (large)	1 23/32"
Clearance inlet valve stem	.015"
Clearance exhaust valve stem	.015"

Valves parallel to center line of cylinder bore.

CYLINDERS.

Overall height of cylinders	10 27/64"
Length of projection in crankcase	3 1/16"
Width of cylinder casting at head over water jacket space	5 7/16"
Width of cylinder casting at barrel over water jacket space	5 1/16"
Length of cylinder casting over water jacket	19 5/32"
Thickness of flange (base)	7/16"
Number of studs per block of four cylinders	20
Diameter of stud	5/16"
Thickness of water jacket wall side and head	5/32"
Thickness of combustion chamber wall	13/64"
Thickness of cylinder barrel, above flange of water jacket	3/16"
Thickness of cylinder barrel, above flange below water jacket	1/32"
Thickness of cylinder barrel below flange	9/64"
Thickness of valve ports	5/32"
Diameter of port at valve, inlet	1 27/64"
Diameter of port at valve, exhaust	2 3/64"
Inlet port at flange (for two cylinders)	2 9/32" x 1⅜"
Exhaust port at flange (for one cylinder)	2 3/64" x 1 31/64"
Number of spark plugs per cylinder	2

PISTON.

Type of piston	Crowned.
Material	Aluminum.
Length of piston	4 1/16"
Length to diameter ratio	.938 : 1
Number of rings per piston; placed two in a groove lower ring beveled acting as wipe ring	4
Position of rings	Above gudgeon pin.
Width of rings	⅛"

Width of land	¼"
Distance from bottom to center of gudgeon pin	2 1/16"
Thickness of head at center	¼"
Thickness of head at edge	⅜"
Thickness of wall at bottom	⅛"

GUDGEON PIN.

Diameter of gudgeon pin	1 5/64'
Thickness of wall	164

CONNECTING ROD.

Type	Plain.
Length between centers	10 7/16"
Ratio length to crank throw	3.313 : 1
Small end bearing	Bronze Bushing.
Outside diameter of bushing	1 3/16"
Length of bushing	2 3/32"
Length of small end of rod	2 3/32"
Outside diameter small end rod at end	1 11/32"
Outside diameter small end rod at center	1½"
Type of section	Eye.
Depth (small end)	1⅛"
Depth (large end)	1 7/32"
Width	⅝"
Thickness of web	5/32"
Thickness of flange (small end)	5/32"
Thickness of flange (large end)	13/64"
Large end bearing	Bronze babbit lined.
Inside diameter bushing	2 3/16"
Outside diameter bushing	2½"
Length	2 9/64"
Thickness of babbit	3/64"

CRANKSHAFT.

Number of crankshafts	2
Number of bearings (Plain) per crankshaft	9
Number of bearings (Ball) per crankshaft	1
Cylinder centers (in block)	4 17/32"
Cylinder centers (between blocks)	7 15/32"
Crank pins, outside diameter	2 3/16"
Inside diameter	1⅛"
Length	2 11/64"
Main Bearings:	
Outside diameter Nos. 1, 2, 3, 4, 6, 7, 8, 9	2 3/16"
Outside diameter No. 5	2⅝"
Inside diameter, Nos. 1, 2, 3, 4, 6, 7, 8, 9	1⅛"
Length, Nos. 1, 2, 3, 4, 6, 7, 8 Bearing Bushings	1 9/16"
Length No. 9 Bearing Bushing	2 19/32"
Length No. 5 Bearing Bushing	1 31/32"
Ball Bearing	Hess-Bright Monarch No. 6211
Crank Webs:	
Width	3 17/32"
Thickness	43/64"
Radius of fillets	3/32"
Weight of one complete shaft, for 8 cylinders, with propeller drive gear bolts and nuts, bevel gear and oil passage shells	96½ lbs.

CAMSHAFT.

Diameter of shaft..1"
Inside diameter..11/16"
Number of bearings..10
Diameter of Bearings Nos. 1 and 10..1"
Diameter of Bearings, Nos. 2, 3, 4, 5, 6, 7, 8, 9..2¼"
Length of Bearings Nos. 1 and 10..1 5/16"
Length of Bearings Nos. 2, 4, 7, 9..⅝"
Length of Bearings Nos. 3 and 8..¾"
Length of Bearings Nos. 5 and 6..27/32"
Width of cam face..5/16"
Number of cams per cylinder..3

CAMSHAFT BEVEL GEAR.

Pitch diameter..3¾"
Number of teeth..30
Pitch 8
Width of face..¾"
Diameter of bolt circle..1 23/32"
Number of bolts..6
Diameter of bolts..5/16"

CAMSHAFT HOUSING.

Material ....Aluminum.

CAMSHAFT DRIVING SHAFT.

Diameter ....¾"
Wall thickness..⅛"

CRANKCASE UPPER HALF.

Material ....Aluminum.
Thickness of wall..3/16" to 5/16"
Thickness of supporting flange..⅜" to 7/16"
Center distance of motor support bolts..24¾"
Number of motor support bolts..12
Diameter of motor support bolts..⅜"
Center to center of crank shafts..10¼"
Height of case..9.055"

CRANKCASE LOWER HALF.

Material ....Aluminum.
Thickness of wall..3/16"

LUBRICATION.

Type..Forced feed and spray.
Type of pump..Rotary Gear.
Number of pumps..One pressure, one scavenging.
Oil pressure..30 #

PRESSURE PUMP.

Number of teeth	7
Pitch	6
Outside diameter	1½"
Width of face	13/16"
Ratio of pump speed to crank shaft speed	1 : 1

SCAVENGING PUMP.

Number of teeth	7
Pitch	6
Outside diameter	1½"
Width of face	1¼"
Ratio of pump speed to crank shaft speed	1 : 1

IGNITION.

Type	Magneto.
Number	4
Make	"Dixie 800"

Firing order 1L, 7R, 5L, 4R, 3L, 8R, 7L, 2R, 4L, 6R, 8L, 1R, 2L, 5R, 6L, 3R

Number of plugs per cylinder	2
Type of plug	Titan A. C.
Advance	38°
Magneto Rotation	Clockwise.

COOLING SYSTEMS.

Type	Water Cooled.
Pump	1 Centrifugal.
Inside diameter of inlet and outlet elbow to cylinders	1½"
Number of inlets	2
Number of outlets	2
Water temperature inlet	150° F.
Water temperature outlet	160° F.

WATER PUMP.

Material	Aluminum.
Inside diameter of inlet	2¼"
Inside diameter of outlet	2 3/16"
Diameter of impeller	5⅞"
Number of blades	8
Ratio pump speed to crankshaft speed	1 : 1

REDUCTION GEARS.

Crank shaft propeller drive gear:

Pitch diameter	5.6"
Pitch	5
Number of teeth	28
Width face	2⅛"
Diameter bolt circle	3 17/32"
Number of bolts	9
Diameter of bolts	7/16"

Propeller shaft gear:

Pitch diameter	8.4"

Pitch	5
Number of teeth	42
Width face	2⅜"
Number of splines	8
Width of splines	.405
Height of splines	.094

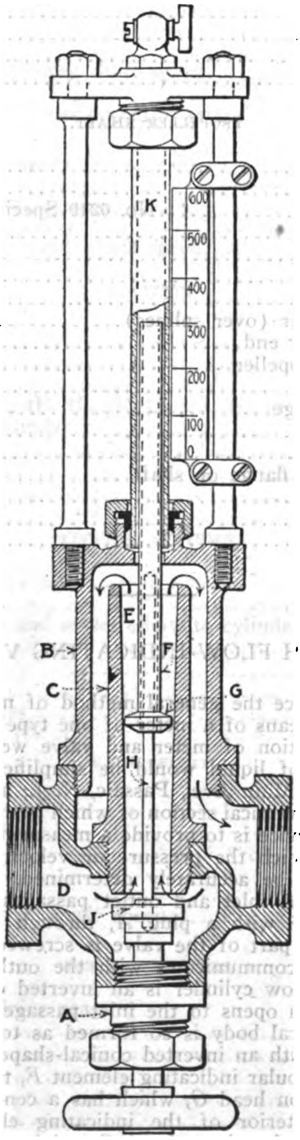
PROPELLER SHAFT.

Number of bearings	3
Type of bearings	Two ball, one plain.
Ball Bearings	No. 6219 Special Width, Hess-Bright.
Plain bearing	Bronze babbitt lined.
Diameter of bearing	3 7/16"
Length of bearing	1½"
Thickness of babbitt	1/32"
Outside diameter at gear (over splines)	3¾"
Outside diameter at rear end	3 5/16"
Outside diameter at propeller	3 31/64"
Inside diameter	2 61/64"
Diameter propeller flange	9¾"
Thickness flange	7/16"
Thickness loose flange	9/32"
length of bearing loose flange on shaft	¾"
Diameter of bolt circle	7⅞"
Diameter of bolts	⅝"
Number of bolts	8

EDLICH FLOW-INDICATING VALVE.

In power-plant practice the general method of measuring fluid passing through a pipe is by means of a meter of one type or another. It is evident that if a combination of meter and valve were made, the pressure or velocity indication of liquid would be simplified. With that idea in mind, Paul Edlich, 134 Main Ave., Passaic, N. J., has patented the Edlich flow-indicating valve, a vertical section of which is shown in the illustration.

The object of this device is to provide a measuring and indicating apparatus for liquids in which the pressure or velocity of the fluid passing through the valve can be accurately determined. The valve casting is provided with the usual inlet and outlet passages. The bottom of the valve body is provided with a plug *A*, through which the valve stem operates. Into the top part of the valve is screwed a hollow cylinder *B*, the interior of which communicates with the outlet of the valve body. Located within the hollow cylinder is an inverted conical-shaped body *C*, the lower end of which opens to the inlet passage *D* in the valve. The lower end of this conical body is so formed as to provide a valve seat. The body *C* is made with an inverted conical-shaped passage *E*, in which is mounted a sliding tubular indicating element *F*, the lower end of which is provided with a piston head *G*, which has a central opening that communicates with the interior of the indicating element, and extending through the opening in the piston head *G* and into the tubular indicating element *F* is the upper end of a guide rod *H*, the lower end of which is inserted into a valve disk *J* and which is operated by means of the handwheel below the plug *A*. A gage glass *K* is placed above the upper end of the hollow body *B*, with an adjoining graduated gage plate, as shown.



SECTION THROUGH INDICATING VALVE

When the valve disc J is closed, the indicating element F will be at its lowest position, or zero. When the valve disc is moved to the open position, a pressure of the fluid against the lower face of the piston head G will force the piston and the indicating element F upward in the gage glass, and as the pressure of the liquid increases, the piston head and the indicating element will move upward in proportion to the amount of pressure, and by virtue of the graduations on the plate the pressure of the liquid against the piston head can be determined.

The liquid, in passing through the conical-shaped passage E, is discharged through the upper end of the passage into the space between the body C and the inner surface of the cylinder body B, as indicated by the arrows, and is then discharged through the discharge passage in the valve body.

THE DIFFICULTIES OF WELDING STEEL BY THE OXY-ACETYLENE PROCESS.*

By B. K. Smith.

The question of steel welding is one of the most important of all the problems which confront the oxy-acetylene welder. A large part of this problem has to do with his preliminary knowledge of steels and his primary habits of handling the blowpipe in steel welding. The steel weld in itself is apparently one of the most easy to obtain, so that some welders are considering steel welding as a subsidiary problem of their trade because their welds look so well; yet steel welding, notably in boiler work, requires the most thought and care, and should be studied not only from the practical point of view, but also theoretically. And what are the difficulties of steel welding? They are: (1) The incorporation of iron oxide; (2) the thermal disturbance in the vicinity of the weld; and (3) the expansion and contraction problems. Therefore one needs to study: (1) The properties of various steels; (2) the phenomena produced during the melting of the metal under the oxy-acetylene torch; and (3) boiler construction.

If one has the above knowledge in a sufficient degree, the welding in itself will be then comparatively easy and successful; but if he has not, and failures occur to him, he will insist that certain operations are unpractical or impossible to perform with the oxy-acetylene blowpipe. This is one of the reasons why certain companies will not trust the apparently good-looking welds, and reserve to themselves the right to sanction or forbid this kind of welding for boiler or pressure work.

In studying the incorporation of iron oxide in the weld, one must know what iron oxide is and what it looks like. Iron oxide is a burnt element, created on the surface of molten steel under the action of the blowpipe and the atmospheric oxygen. It appears as white veins and foaming spots. Iron oxide will dissolve into the molten steel at the rate of a little higher than 1 per cent, and in this reaction it may destroy the carbon elements, with the result of a decrease in the strength and elasticity of the weld.

The best remedy to reduce oxidation is to reduce the oxygen pressure as low as possible—just enough to produce a free and soft flame that will produce a continuous melting without running over the edge. This operation depends largely on one's blow-pipe and the flame maintained throughout the weld. With a proper flame, one will notice the flow of the metal clean and regular, but with a too rigid flame which requires more

*Abstract of a paper read before the North-Western Welding Association, Minneapolis, Minn.

oxygen one will notice white veins and foamy spots flowing in the molten metal. These are streaks of oxide of iron, and they will dissolve and create new ones with the progress of the weld. The molten metal is swept rather than laid in clean layers, and in some cases the metal is adhered instead of welded. Oxide of iron cannot be altogether eliminated from any kind of weld so long as we cannot protect them from the effects of the atmosphere, but it can be so decreased from the interior of the weld that its contents would do no harm to the strength of the weld.

Some good and experienced welders who have been welding for five or six years, but who have met with some failures in steel welding, will not admit that the main cause of failure is the improper regulation of the flame, because to do so would be the same as for a college student to be compelled to return to the grades to learn his A B C's. Most of our welders have previously mastered some other craft and have begun welding on important work without any preliminary instruction. In other words, they began at the college end without having acquired the necessary underlying primary and high-school education.

What would a boilermaker think if a novice tried to put in a set of flues? Or a machinist if an apprentice attempted to make a set of dies? Yet the same boilermaker or machinist looks upon the welding blowpipe as merely a tool of his trade, and thinks that all he needs to do is to go ahead and weld with it.

In studying the question of the thermal disturbance produced by the gas flame in the metal when it is raised to a fusing temperature we must not forget that it is heat which has led to the creation of many new steels, and while heat can be employed to increase the life of metals it can also reduce and destroy the life and strength of metals. Therefore, it remains for us to study carefully how we shall best make use of it.

It is a well-known fact that the structure of any gas weld at the welding line, on a boiler plate, for instance, is cast metal. This metal is much lower in elongation than the boiler plate. Yet the structure of the boiler plate was originally at the mill a cast metal or something similar to the metal in the weld. But, it has been refined by the operations of heat and mechanical treatment to a strong metal that we call boiler plate. The question is this: Can we employ the heating agent with the aid of the gas blowpipe to refine the grain at the weld and its vicinity? My answer is, "Yes," provided the operator knows the physical and the chemical properties of boiler steel.

There is no such thing as not being able to take care of expansion and contraction in steel welding. But in boiler welding one will find this a problem which is difficult to solve unless one has a full knowledge of the construction of boilers.

The effects of expansion and contraction are little feared by some boiler welders, because the metal to be welded possesses the property of elongation. My advice is that no welder should depend upon the metal giving, thinking that a little strain will not hurt the weld, for two reasons. One reason is that in almost all cases an intelligent welder can find a way to take care of the expansion and contraction. The second reason is that no human being can ever measure or know how much of a strain he has left in the metal. It may be very little or it may be up to the breaking point. Such a weld may crack during its own progress or a few days after, or even six months after completion; at any rate, the welder is not excusable, and the failure will not only condemn the operator, but the process and the whole oxy-acetylene welding industry.

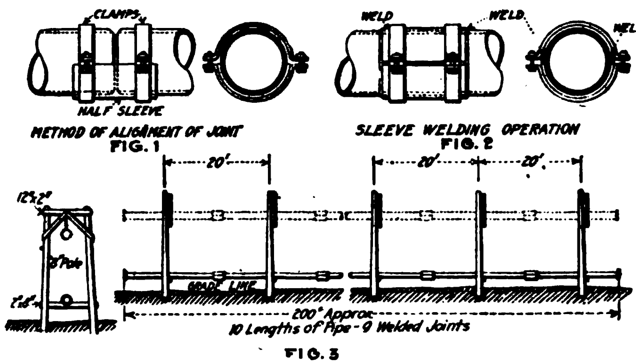
There is one point that an oxy-acetylene operator must bear in mind, and this is to make a strong weld. Economy should be considered as a secondary principle. While there must not be any useless waste in time or material, yet we must not forget that a defective weld is a waste of time and material.—" Mechanical World."

ELECTRICALLY WELDED STEAM LINES.*

BY H. R. WOODROW, ASSISTANT CHIEF ELECTRICAL ENGINEER, NEW YORK EDISON CO.

In the construction of the United States Government Explosives Plant "C" at Nitro, W. Va., something over ten miles of steam lines carrying 190 pounds pressure were required. Of this about five miles were 12 inches diameter or over. At the time the construction of this plant was undertaken, in the spring of 1918, the available supply of large extra-heavy flanges was very small and it was doubtful if a sufficient quantity could be obtained within the time available for completing the plant. Therefore in order to finish the work in time the Thomas E. Murray Engineering Corp., which was handling the power-section engineering work, recommended, and the War Department adopted, the use of arc welding for the connection of all the high-pressure steam and feed-water lines 6 inches in diameter and over in the plant. At the time the armistice was signed over one thousand of these joints had been completed and were in service.

All the outside mains are suspended from timber bents, at a height of approximately ten feet about the ground, as shown in Fig. 3. The welding operation was performed directly under the bents and the section of welded-joint pipe, after being tested, was raised in place by means of chain blocks.



FIGS. 1 TO 3.—METHOD OF WELDING JOINTS AND INSTALLING STEAM LINE.

The length of welded-joint pipe so handled averaged approximately 200 feet and contained ten sections of mill-length pipe and nine welded joints, the two end sections having a standard Van Stone flange for connecting with fittings or valves in the completed line.

The type of welded joint used is a combination butt and sleeve joint, as indicated in Figs. 1 and 2. The butt ends of adjoining sections were scarfed off at an angle of about 45 degrees with the axis of the pipe and butt-welded. The weld was then chipped flush with the pipe surface, and a hydrostatic test of from 400 to 600 pounds was applied to the completed length. After inspection, the reinforcing sleeve was applied. The

*Presented in the discussion on electric welding at a joint meeting of the American Institute of Electrical Engineers and the American Institute of Mining Engineers, held during the Midwinter Convention of the American Institute of Electrical Engineers, New York, N. Y., Feb. 19, 1919.

sleeves are made in halves and are die-formed of metal equal in thickness to, and of a length approximately equal in diameter to that of the pipe to which they are applied.

The welding was performed by the pencil-arc process, the current employed being 110-volt alternating, which was obtained primarily from the general supply system for the plant, carrying 6,600-volt three-phase current led through transformers and secondary lines to the welding reactances, furnishing the controlling medium required for the welding operation.

The electrodes used in the butt-welding operation were slag-coated and were of the same character as those extensively used in ship welding. The welding of the sleeves was performed with a low-carbon steel bare wire. The butt weld consisted of two runs of metal, the first of No. 10 and the second of No. 8 wire, the first run being carefully cleaned free of slag, first by hammering and finally by scrubbing with a wire brush until the metal showed clean and bright before applying the second run. The sleeve weld consisted of two longitudinal welds joining the two halves of the sleeve, and a weld on each end of the sleeve completely joining the sleeve to the pipe (See Fig. 2).

The application of the sleeve was determined upon for structural reasons, as the stresses likely to be encountered in an installation of this character are practically indeterminate.

One of the problems met with in handling pipe in such long lengths is the question of accurate alignment, which was accomplished in this case in the following manner: The proper level above the ground for the pipe was determined by its accessibility to the welder, and this point was marked on each leg of each pipe-carrying bent, and timber joists were securely fastened between the legs of the bents at the proper level, as in Fig. 3. The pipe lengths, approximately 20 feet long, were laid on these joists, and as the spacing of the bents along the axis of the pipe was 20 feet, this brought a joint between each pair of bents. The ends of the individual pipe lengths were then brought within about 1/10 of an inch of contact and lined up and clamped securely in position by means of a half-sleeve applied to the under side of the pipe and fastened by straps to each end of each pipe section (See Fig. 1).

This left the upper half of the butt joint ready for the first run of the weld. When one-half of the first run was completed on each joint, the clamps and the sleeve were removed and the entire 200-foot length of pipe was rolled on the joists until the other half of the butt joint was uppermost, and the first run was then completed. In consequence of the fact that it was possible to turn the pipe at will thereafter, the second run of the weld was made continuous.

It takes one welder about three hours to completely butt-weld and sleeve a 12-inch joint. The time required for other sizes would be in proportion to their diameter.

I feel that the use of the electric welding is going to be a very important feature in the development of power stations where the tendency is going to higher steam pressures, in the neighborhood of 600 pounds, as welded joints can unquestionably be made stronger than any other connection and are absolutely leakproof.—" Power."

INSPECTING A SURRENDERED GERMAN DESTROYER "S" TYPE.

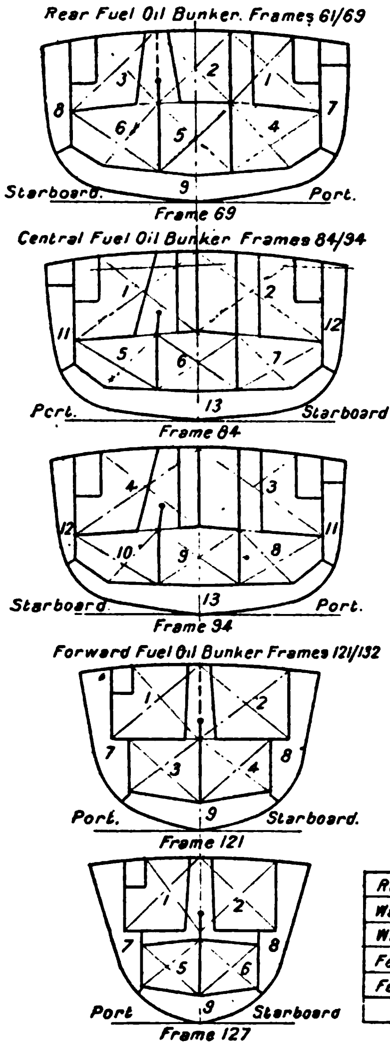
The following notes describe the impressions formed on inspecting a German "S" type destroyer built in 1914. The boat was supposed to be efficient and ready for sea in all respects, but personally the writer would

not like to take her to sea for an extended trip without an extensive refit. Considering the hull first, the hull and decks were very dirty and appeared in a very neglected condition. The bridge was certainly an improvement on the bridge of the British 1914 boats, but this was the sole improvement one could see. She was fitted with two rudders, one fore and one aft, the forward one being housed when not in use, being lifted up by gearing. Apparently the forward rudder was used in a crowded fairway or when entering and leaving harbor. Steering was by chain transmission, a very defective method according to our standards. The steering engines, two in number, were housed on the upper deck, one almost right aft and the other on the mess deck forward, a little forward of the bridge. The steering engines were vertical, much cramped, and ungetatable for necessary examination and repairs.

In addition to the main torpedo tubes, two smaller ones were fitted, one on each side at the break of the forecastle. These appeared to have no protection from heavy seas in bad weather. The storerooms, notably the engineer's storeroom, were extremely cramped and small and very dirty. The magazine seemed far too small for a boat of her armament. To both storerooms and magazines the means of access were very inadequate. The warrant officers were provided with a separate mess aft, and both this mess and the wardroom were cramped and looked most uncomfortable. The cabins opened from the wardroom and were very small and ill-found. The wardroom pantry and galleys seemed quite inadequate for the number of officers and men carried, and were in a very bad condition.

Her oil tanks were situated amidships and forward, and were carried right across the ship and up to the upper deck. This method is undoubtedly very inferior to the British practice, danger from fire, explosion and gunfire being materially increased. The manholes for the oil tanks opened to the upper deck, the only advantage in this being a little time saved when "oiling boat." The deck space was very restricted, the spacing and arrangement of the various fittings being much inferior to ours. The telegraphs were by wire transmission, both from the bridge to the engine room and from the engine room to the boiler rooms. This method is very bad and in our service is obsolete. The general finish of the hull construction throughout the vessel was inferior to ours, and she was much more lightly constructed. It is not considered that in weather similar to that in which our flotillas maneuver she would stand half the knocking about and strain our boats are constantly subjected to. The freeboard is low, and it would be interesting to follow their behavior in a bad seaway.

The following points were noted as regards the propelling machinery in the short time available. The boat is twin-screw, driven by turbines, and has two engine rooms, one forward of the other. The turbine speed is reduced to propeller speed by a method of transmission the details of which could not be ascertained. The condensers are on the platforms, not underslung, which method it is considered would add greatly to the boat's efficiency, as well as giving much more space could it be adopted. The auxiliary engines were very cramped and had a most uncared-for appearance. The dynamos were small 80-volt machines, and two small evaporators, which at a casual glance appeared to be quite inadequate for the boat's requirements, were fitted; at a rough guess their output would not be more than 24 tons of fresh water per day. The starting platforms were most restricted, and the starting wheels large and cumbersome. Engine-room telegraphs were roughly finished and, as mentioned, transmission was by wire. The ventilating and exhaust engine-room fans were of the horizontal type, and appeared of ample size to do the work required. The auxiliary feed pumps were situated in the engine rooms. The various gages seemed to be in a most inefficient condition and more were required for general efficiency. The electrical apparatus in the engine rooms



1. Port, upper
2. Central, upper
3. Starboard, upper
4. Port, lower
5. Central, lower
6. Starboard, lower
7 Side tank, port
8 Side tank, starboard
9. Double bottom
1. Port, upper
2. Starboard, upper
3. Port, upper
4 Starboard, upper
5. Port, lower
6. Central, lower
7. Starboard, lower
8 Port, lower
9. Central, lower
10 Starboard lower
11. Side tank, port
12. Side tank, starboard
13. Double bottom
1. Port, upper
2. Starboard, upper
3. Port, lower
4 Starboard, lower
5. Port, lower
6. Starboard, lower
7. Side Tank, port
8. Side tank, starboard
9. Double bottom
Total (cubic metres)

Amount of Feed Water.

Reserve tank, after engine room
Waste tank, after engine room
Waste tank, forward engine room
Feed water tank I
Feed water tank II
Total (cubic metres)

DIAGRAM OF OIL FUEL TANKS OF A GERMAN TORPEDO BOAT DESTROYER, "S" CLASS.

A daily record is kept of amount of oil fuel in each tank, and logged in spaces at side of diagram.

appeared up-to-date and efficient, and there seemed a good supply of electrical stores and fittings. The general condition of both engine rooms and main and auxiliary machinery was dirty in the extreme, and a total lack of care seemed to have been expended on its maintenance. Access to the engine rooms was very restricted, and several of the important auxiliaries were very inaccessible for examination and repair.

Three small-tube water-tube boilers were installed, two back-to-back in the after boiler room and the other in a separate compartment, the boilers being numbered from forward. The boiler rooms were in far worse condition even than the engine rooms and were in a deplorable state. In a long and varied experience the writer has never seen such dirty boiler rooms—a tramp steamer after a long, rough voyage would put them to shame. The chief feature of the boiler rooms is the lack of space both at the sides and at the fronts. To one used to British design it seemed impossible that efficiency could be maintained under such conditions. The sprayers for the oil fuel were much larger and more cumbersome than in our service, and the adjustment for the oil output was of a very crude nature, consisting of a lever and ball arrangement. Eight sprayers were fitted to each boiler. The forced-draft fans were of the horizontal type and when inspected with only auxiliary machinery running 1½ inches of air pressure was maintained. There appeared to be one oil-fuel pump to each boiler and one main feed pump. These latter seemed to be a bastard Weir design. Unfortunately, time did not permit of an extended examination of the boilers, uptakes and funnels. The runs of piping in both engine and boiler rooms seemed very complicated.

From a table found in the engine room the oil fuel consumption for the various speeds appeared to be as follows: 10 knots, 1.34 tons per hour; 12 knots, 1.54 tons per hour; 15 knots, 2.18 tons per hour; 20 knots, 4.55 tons per hour; 30 knots, 16.06 tons per hour. This last figure compares very unfavorably with the consumption of our 1914 destroyers. Another feature which was of interest was that there were no forced-lubrication auxiliaries, which is most extraordinary in this type of vessel As far as one could gather, a speed of 33 knots had been obtained, but no data was available to verify this. The oil consumption and speed trials, I should imagine, had been carried out in smooth water at light draft, and with the machinery in an absolutely efficient condition. Another feature was that there were no special appliances fitted to the boilers for the purpose of making "smoke screens," and the only method they could adopt seemed to be by reducing the speed of the forced-draft fans, keeping the same quantity of liquid fuel passing through the sprayers.

It is to be hoped that at a future date more data will be available, especially as regards oil-fuel consumption, lubricating oil used, speed actually obtained and method of transmission.—"Shipbuilding and Shipping Record."

BOOK REVIEWS.

HEAT. By E. M. SHEALY. Published by MCGRAW-HILL BOOK CO., New York. 262 pp., 110 ill.

STEAM BOILERS. By E. M. SHEALY. Published by MC-GRAW-HILL BOOK CO., New York. 356 pp., 185 ill.

STEAM ENGINES. By E. M. SHEALY. Published by MC-GRAW-HILL BOOK CO., New York. 290 pp., 173 ill.

The above is a set of text books for correspondence students in the University of Wisconsin Extension Division. The set forms a very useful library for operating engineers and firemen, written in as non-technical style as possible.

" Heat" first treats the fundamental laws governing generation, transfer and transformation of heat and illustrates these laws by familiar examples in such manner as to create and hold the reader's interest. These laws are followed by a simple treatment of steam and the various media for refrigerating machinery. The principles of steam and gas engines, refrigerating machines and air compressors are treated in a practical manner adapted to the needs of the student and operator.

" Steam Boilers" is written for the practical man—the fireman, the man in charge. Fireroom equipment and the operation of boilers is stressed, little space being devoted to design. Fuels and the theory of combustion prepares the student for the sections on firing and smokeless combustion.

" Steam Engines" treats of the fundamental principles underlying the operation of the steam engine in such a manner as to enlist the interest of the average operating engineer. Of the nineteen chapters, six deal with valves and valve gear; the reason for this large proportion is given by the author because

his experience shows that this is a subject least known by operating engineers. Chapters on condensing apparatus and lubrication, engine testing and governing distinguish this book from the usual run of books in this field.

SHIP STABILITY AND TRIM. By PERCY A. HILLHOUSE, was reviewed in the last issue of this Journal. The U. S. publishers are D. VAN NOSTRAND Co., New York, N. Y. Price, $4.50.

THE MARINE STEAM TURBINE, 5th ed.. By J. W. M. SOTHERN. Published by D. VAN NOSTRAND Co., New York, N. Y. 756 pp., about 700 ill.

This new edition (5th) of this well-known work has been enlarged by more than 200 pages and 350 illustrations over the 4th edition. The material of the previous edition has been thoroughly revised, and the new material incorporated includes: Impulse Turbines, Turbo-generators, Double Reduction Gearing, Marine Electric Propulsion, Michel Thrusts, Feed Regulators, Torsion Meters, recent developments in turbine practice, and a section has been added on the practical operation of marine turbines, containing much valuable data from actual tests and trials. This latter section should be of particular interest and value to the operating engineer.

Among other new material there is included a reprint of Mr. Roland S. Portham's paper, read before the Institute of Engineers and Shipbuilders in Scotland, on the " Ljungstrom turbine and its application to marine propulsion," with a full description of the installation of Ljungstrom Turbine Machinery on the S. S. *Wulsty Castle*. (See " Proc. A. S. N. E.," Nov., 1918.)

The unusual profusion of illustrations makes this book most pleasing reading and adds very materially to its value as an educational work. No pains have been spared in this respect. An excellent work for every engineer's book shelf.

RECRUIT MANUAL. By GEORGE C. THORPE, Colonel, U. S. M. C. Published by J. B. LIPPINCOTT Co., Philadelphia. 168 pp.

This manual is intended to excite the recruit's interest by answering the many questions propounded in the beginning. This is what the author terms "the psychological moment to seize upon his interest and to fix a spell upon him that will keep his interest going." General instructions to the recruit, simple methods of learning signalling, the manual of arms, the rifle and how to handle it, first aid measures, extracts from "Articles for Government of the Navy," and a military glossary are some of the features of this work. Pocket size, it is the best recruit manual we have seen.

MESS MANAGEMENT, by COLONEL WILLIAM E. DUNN, 344th Field Artillery. J. B. LIPPINCOTT Co. 16mo. Flexible cloth, $1.00 net.

This small volume, which can be carried in the pocket, is a guide for mess officers on how to organize and how to supervise the operation of the mess, so as to insure the men being well fed on the Government money allowance for food. It contains detailed directions on how to keep mess accounts, how to plan menus, sample menus being included, and how to plan purchases.

When it is considered that an army is said to fight on its stomach—for it is certain that no body of ill-fed troops can possibly be effective for any length of time—the importance of making the best possible use of the food-money allowance is seen to be a vital need. The secret of running a good mess is to put plenty of effective headwork into it. Colonel Dunn's practical manual will aid to make this headwork effective.

GEORGE WESTINGHOUSE. His Life and Achievements. By FRANCIS E. LEUPP. LITTLE, BROWN AND Co., Boston, 1918. Cloth, 6 x 9 in., 304 pp., 5 pl., 6 portraits. $3.

Mr. Westinghouse counted many readers of this Journal among his friends and this volume will hold special interest for them. It deals with the personal side of Mr. Westinghouse's career. Mr. Leupp frankly confesses that the mission of this volume is "simply human." No one could read this biography of one of the most prominent men of his time in the industrial world without receiving inspiration therefrom. It points a strong moral for the young man of today.

Compiled without the aid of diaries or files of personal correspondence, and relying upon the memory of associates, newspaper and magazine articles and the like for data, Mr. Leupp deserves great credit for giving us this true human picture of the wonderful character of this man.

Engineer, inventor, administrator, business man and father to his large industrial family, Mr. Westinghouse worked so successfully in so many lines that this work should have its companion piece dealing with the technical side of his career. Who will write it?

NAVAL POWER IN THE WAR. By C. C. GILL, Commander, U. S. Navy. GEORGE H. DORAN CO., New York, N. Y. 302 pp., 11 maps and diagrams, 9 ill. $1.50 net.

Commander Gill performed active service during the war in the Cruiser and Transport Service and can write as an authority on this subject. He has put the story of the Naval operations of the war to the date of signing the armistice into excellent brief shape.

The opening naval activities of the war, the Battle of Jutland and other major actions, the results of such actions, submarine and anti-submarine warfare, the part this country played in the naval activities, the transport and convoy system, all lead to the concluding section which treats of the Naval Lessons of the War.

The whole subject is graphically treated. This book is approved for use as a text book by the Academic Board of the U. S. Naval Academy.

Practical Ship Production. By A. W. Carmichael, Lieutenant Commander, C. C., U. S. Navy. Published by McGraw-Hill Book Co., New York. 250 pp., 101 ill.

This book presents in convenient form the most important general principles of ship design and construction. The treatment is practical rather than theoretical and is intended to fit the practical engineer or production man, who is well grounded in mathematics and the mechanical processes, to turn his talents to shipbuilding.

To workmen in the shipyards this volume is specially recommended. Its contents include the requirements and general descriptions of ships, structural members and design of ships, shipyards, preliminary steps in ship construction, and the building of ships. Within its limitations, quite the best book of the year in this field.

ASSOCIATION NOTES.

ADDRESSES.

To ensure delivery of your Journal promptly, the necessity of your keeping the Secretary advised of your address at all times cannot too strongly be urged.

The following members and associates have joined the Society since the publication of the last Journal:

MEMBERS.

Astoria, Philip, Lieut., U. S. N.
Bandura, Oscar F., Lieut., U. S. N.
Durgin, Calvin T., Lieut., U. S. N.
Grommer, J. G. B., Lieut. Commander, U. S. N.
Hobby, Thomas F., Lieut., U. S. N.
Kershaw, Herbert E., Lieut., U. S. N.
Mayville, Armand, Lieut. (CC), U. S. N.
Owen, W. C., Commander, U. S. N.
Usina, M. N., Capt. of Engineers, C. G.

ASSOCIATES.

Akimoff, N. W., 1013 Harrison Building, Philadelphia, Pa.
Blankenhorn, George S., 215 Drexel Building, Philadelphia, Pa.
Borum, J. V., 125 Oak St., Jacksonville, Fla.
Burch, Charles H., Lieut., U. S. N. R. F.
Chapman, Lawrence B., Electric Boat Co., Groton, Conn.
Collett, Eugene, 1134 50th St., Brooklyn, N. Y.
Cooper, Lewis M., Lieut. U. S. N. R. F.
Eriksen, O. E., 1825 Market St., Jacksonville, Fla.
Hawthorne, W. G., 115 Broadway, New York City.
Hocken, Barry, Engineer Commander, R. British Navy.

Hoyer, K., care Newport News Shipbuilding and Dry Dock Co., Newport News, Va.

Hubbard, William E., Lieut., U. S. N. R. F.

Hymans, F., Glen Ridge, N. J.

Knight, A. Raymond, South Eliot, Me.

McGarvey, Joseph T., care American and British Manufacturing Co., Bridgeport, Conn.

Petrie, G. Rollo, P. O. Box 952, Newport News, Va.

Phillips, H. M., 1211 Fern St., N. W., Washington, D. C.

Seward, Herbert L., 966 Townsend Ave., New Haven, Conn.

Slocum, Stephen E., Prof., University of Cincinnati, Cincinnati, Ohio.

Smith, Alfred O., Lieut., U. S. N. R. F.

Westwater, A., P. O. Box 237, Everett, Wash.

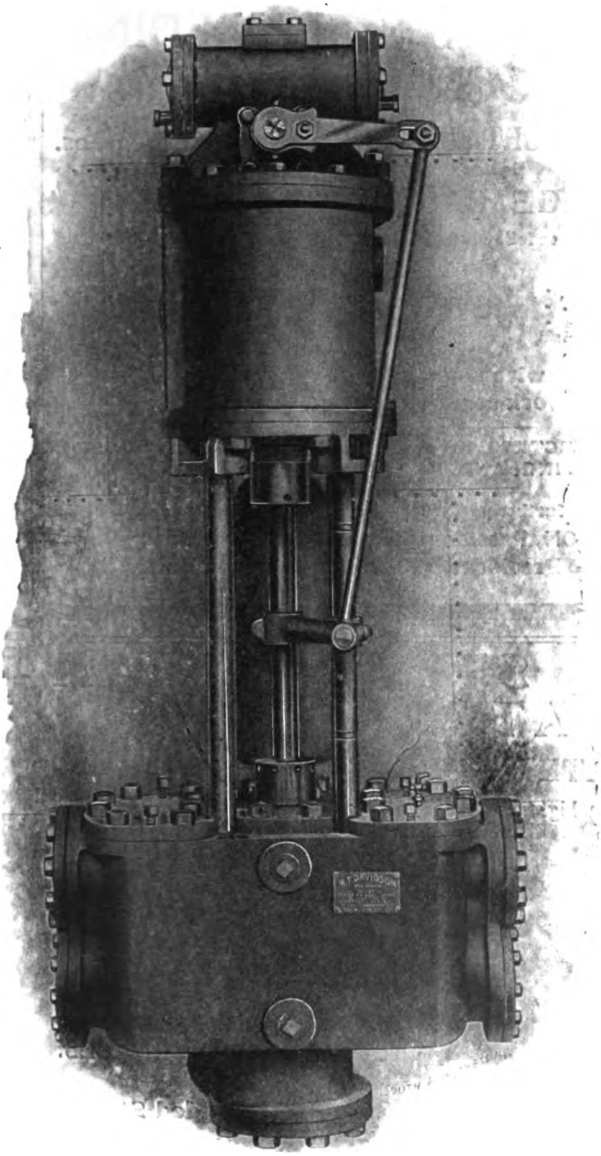
Yarrow, Harrold E., Yarrow & Co., Ltd., Scotstoun, Glasgow, Scotland.

ANNOUNCEMENT.

THE TERRY STEAM TURBINE COMPANY makes the deferred announcement that since the death of Mr. Charles E. Hague in April, 1917, their interests in the Philadelphia District have been under the supervision of Mr. V. L. Sanderson, with headquarters at 605 Widener Building.

They further announce that Mr. James H. Weir has recently become associated with them as assistant to Mr. Sanderson in that district.

They earnestly solicit requests for engineering advice along all lines of turbine application.



DAVIDSON STEAM PUMPS

FOR ALL SITUATIONS.

CONDENSERS,
EVAPORATING
and
DISTILLING
APPARATUS.

U. S. S. "Connecticut",
"Washington",
"St. Louis",
"Denver",
"Chattanooga",
"Bancroft",
"Baltimore",
"Cleveland",
"Galveston",
"Iris",
"Rainbow",
"Arkansas",
"Bagley",
"Dale",
"Rodgers",
"Stewart",
"Worden",
"Winslow",
"Gloucester",
&c., &c., &c.

M. T. DAVIDSON COMPANY

43-53 Keap Street,
BROOKLYN, NEW YORK.

NEWPORT NEWS SHIPBUILDING and DRY DOCK COMPANY.

WORKS LOCATED AT **NEWPORT NEWS, VA.,** ON HAMPTON ROADS.

Equipped with

THREE LARGE BASIN DRY DOCKS

of the following dimensions:

	DOCK No. 1.	DOCK No. 2.	DOCK No. 3.
Length on top,	610 feet	827 feet	587 feet
Width on top,	130 "	162 "	147 "
Width on bottom,	50 "	80 "	62 "
Draught of water over sill,	25 "	30 "	24 "

Shops are equipped with modern machinery capable of doing the largest work required in ship construction.

TOOLS DRIVEN BY ELECTRICITY AND COMPRESSED AIR USED IN CONSTRUCTING AND REPAIRING VESSELS.

For further particulars address

H. L. FERGUSON, President, Newport News, Va.,
F. P. PALEN, Vice President, 233 Broadway, New York.

WELDED STEEL STEAM and WATER DRUMS

for WATER-TUBE BOILERS.

Embodying Strength with Lightness and freedom from LEAKAGES Incident to Riveted Structures.

MORISON SUSPENSION FURNACES.

FOX CORRUGATED FURNACES.

Welded Steel Tanks, Receivers, etc.

The CONTINENTAL IRON WORKS,

West and Calyer Sts., BOROUGH OF BROOKLYN, N. Y.,

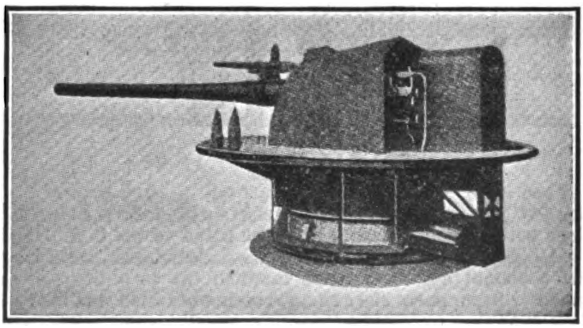
ESTABLISHED 1859
INCORPORATED 1887

Greenpoint Ferry from East 23d Street, New York.

BETHLEHEM STEEL CO.

SOUTH BETHLEHEM, PA. 25 Victoria Street, LONDON

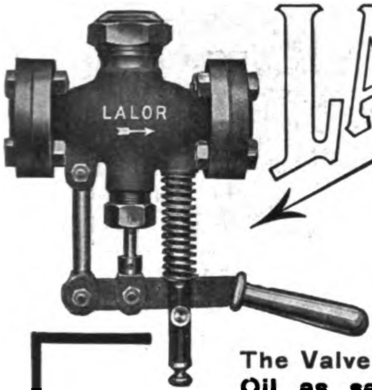
Naval, Field and Coast Defense GUNS and MOUNTS Armor Turrets Projectiles



9."2 Coast Defense Gun with Sub-calibre Equipment, on Barbette Mount.

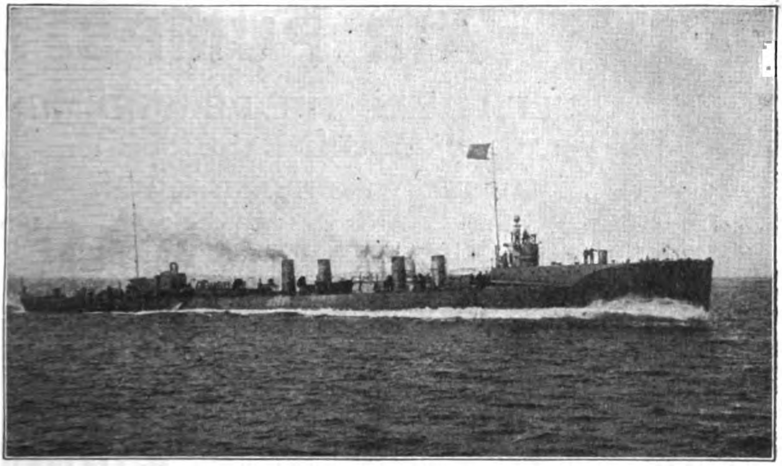
Forgings Castings Shafting Rails and Structural Steel

We are continuously manufacturing ORDNANCE MATERIAL for the U. S. Army, U. S. Navy and Foreign Governments


LALOR

BATH IRON WORKS, LTD.

BATH, MAINE.



BUILDERS OF DESTROYERS, FAST YACHTS AND PASSENGER STEAMERS. NORMAND WATER TUBE BOILERS. PARSONS MARINE STEAM TURBINES. TWENTY-FIVE DESTROYERS BUILT OR BUILDING FOR THE UNITED STATES NAVY, ALL HAVING SPEEDS BETWEEN THIRTY KNOTS AND THIRTY-SEVEN KNOTS.

AFLOAT AS WELL AS ASHORE, JOHNS-MANVILLE Products are designed to save power, fuel and equipment. Johns-Manville Service puts at your disposal competent engineers whose recommendations, after a study of your requirements, secure for you the full benefit of the efficiency in Johns-Manville Products. The following partial list may be suggestive:

FOR THE POWER PLANT:

- Packings and Gaskets, sheet and mechanical
- Pipe and Boiler Insulations
- Refractory and Insulating Cements
- Cork Materials
- Steam Traps
- Brake Blocks and Lining
- Tachometers

FOR THE ELECTRICAL PLANT:

- Ebony and Transite Asbestos Wood
- Hard Fibre and Moulded Insulations
- Fuses and Meter Protective Devices
- FIRE EXTINGUISHERS

Audiffren-Singrun Refrigerating Machines

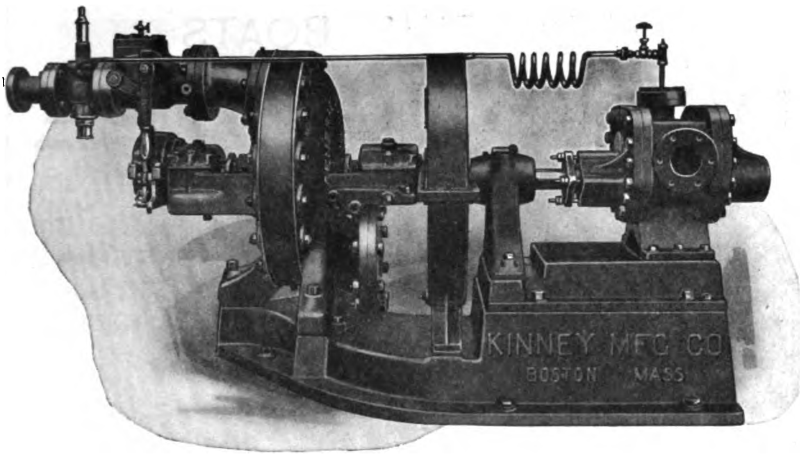
JOHNS-MANVILLE SERVICE

COVERS THE CONTINENT

H. W. JOHNS-MANVILLE CO.

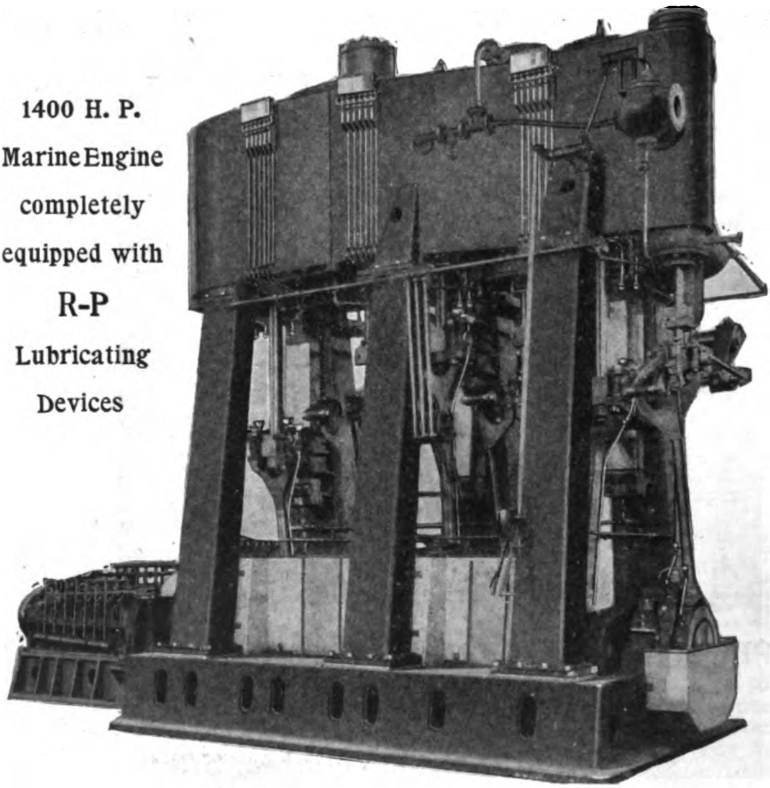
NEW YORK CITY

10 Factories — Branches in 63 large cities


KINNEY MFG CO
BOSTON MASS

LUBRICATION

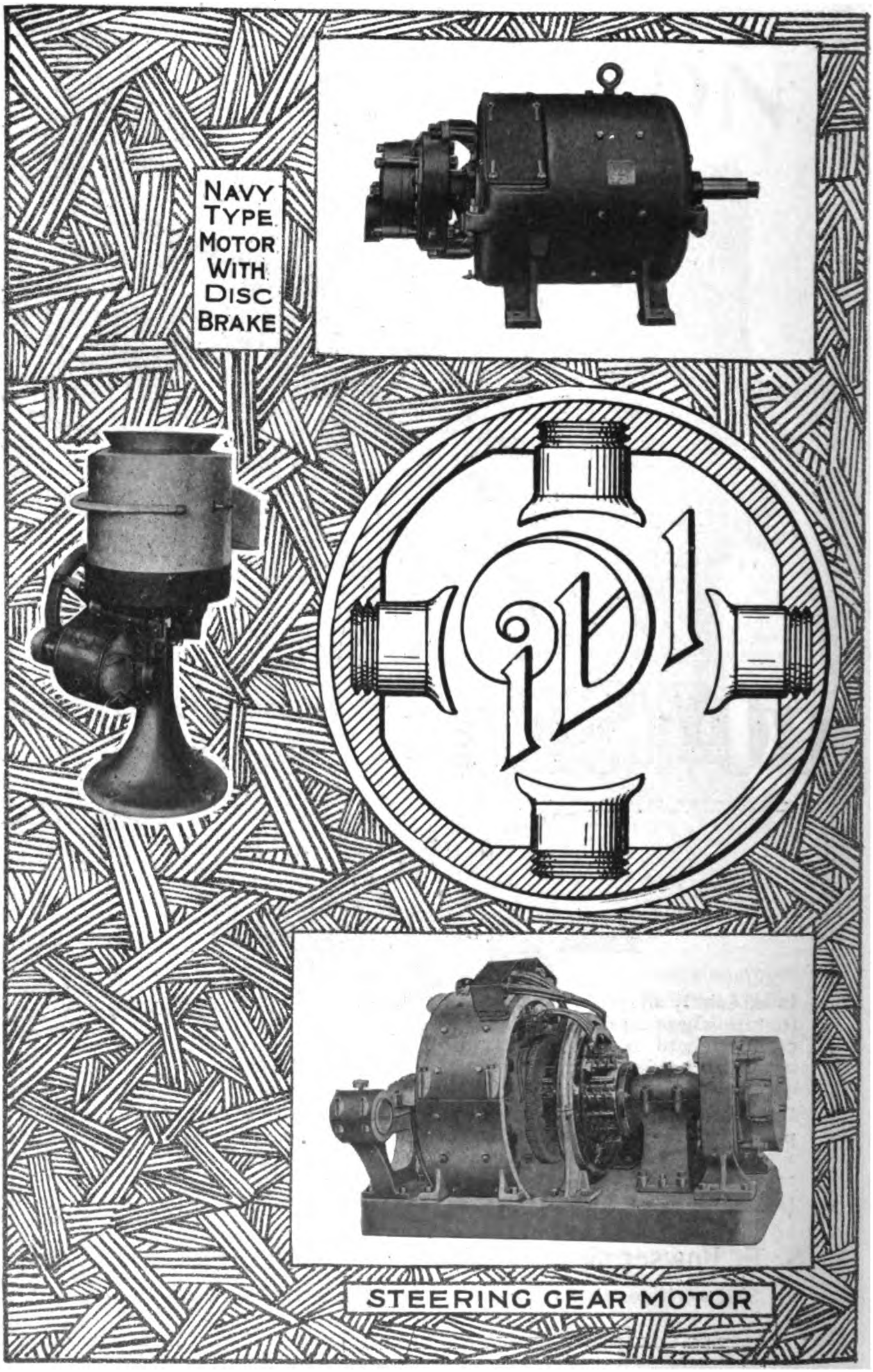
1400 H. P.
Marine Engine
completely
equipped with
R-P
Lubricating
Devices

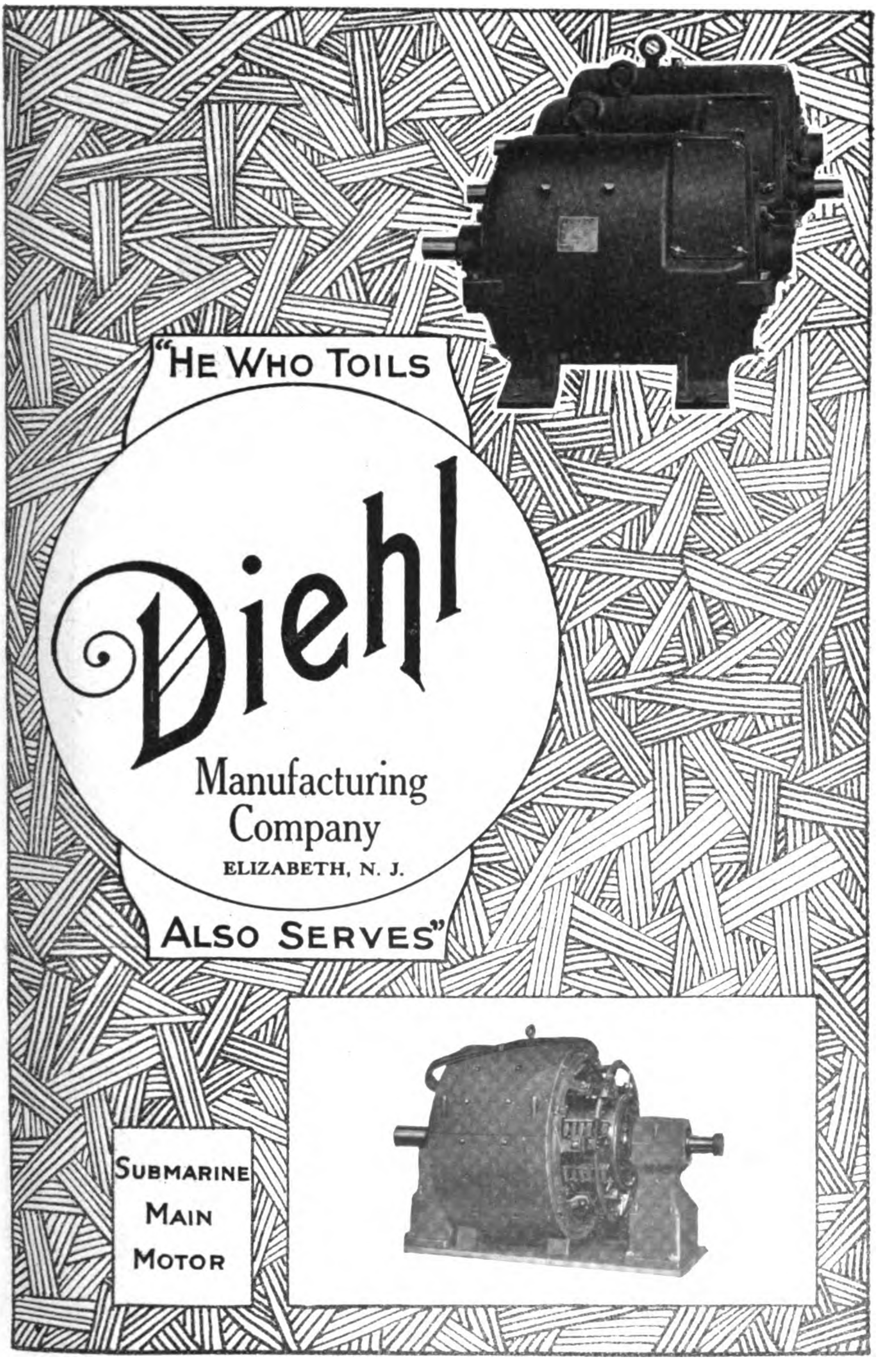


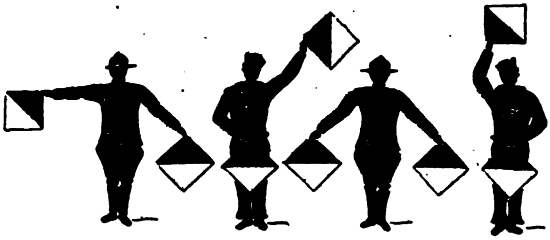
During the war hundreds of Marine Engines were completely equipped with R-P Standardized Lubricating Devices, consisting of Multifeed Wick Oiler Boxes—Crank, Wrist Pin and Link Block Oil Cups—Pipe and Drip Oil Cups—Guide Combs, etc.

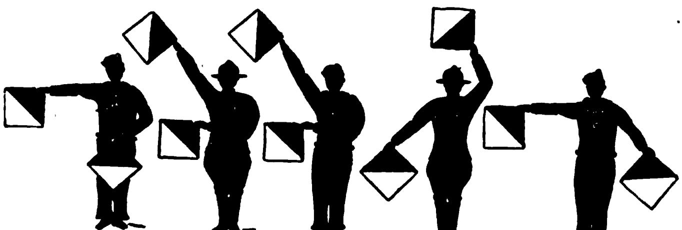
R-P Standardize Oiling Devices for Marine Engines are described and illustrated in Engineering Publication A-26. A copy is yours for the asking. Will you drop us a postal for it today?

THE RICHARDSON-PHENIX CO.
LUBRICATION ENGINEERS AND MANUFACTURERS
WORKS: 114 RESERVOIR AVE. MILWAUKEE, WIS.


NAVY
TYPE
MOTOR
WITH
DISC
BRAKE
STEERING GEAR MOTOR


"He Who Toils
Diehl
Manufacturing
Company
ELIZABETH, N. J.
Also Serves"
Submarine
Main
Motor





VOL. XXXI. NO. 2.

JOURNAL

OF THE

AMERICAN SOCIETY

OF

NAVAL ENGINEERS.

PUBLISHED QUARTERLY BY THE SOCIETY, FOR THE ADVANCEMENT OF NAVAL ENGINEERING.

MAY, 1919.

WASHINGTON, D. C.

R. BERESFORD, PRINTER, 605 F STREET, N. W.

Entered at the Post Office at Washington, D. C., as Second Class Matter.—Acceptance for mailing at special rate of postage provided for in Sec. 1103, Act. of Oct. 3, 1917, authorized on July 3, 1918.

OFFICERS OF THE SOCIETY.

President:

Rear Admiral H. P. NORTON, U. S. Navy.

Secretary-Treasurer:

Lieutenant Commander F. W. STERLING, U. S. Navy (Retired).

Council:

Rear Admiral H. P. NORTON, U. S. Navy.
Rear Admiral T. W. KINKAID, U. S. Navy.
Captain ROBERT STOCKER, (C. C.) U. S. Navy.
Commander S. M. ROBINSON, U. S. Navy.
Commander W. W. WHITE, U. S. Navy (Retired).
Captain of Engineers M. W. TORBET, C. G.
Lieutenant Commander F. W. STERLING, U. S. Navy (Retired).

SUBSCRIPTION RATES.

The subscription price of the JOURNAL, postpaid to the United States and possessions, Canada, Cuba and Mexico, is $5.00. Single copies, $1.30. To other countries in the Postal Union, $5.40. Single copies $1.35. All subscriptions are payable in advance.

Make checks, drafts and postal orders payable to

AMERICAN SOCIETY OF NAVAL ENGINEERS.

Advertising rates will be furnished on application.

It is earnestly requested that prompt information be given of changes in address, or of failure to receive the JOURNAL.

Address all communications for the Society to

AMERICAN SOCIETY OF NAVAL ENGINEERS,
Navy Department, Washington, D. C.

CONTENTS.

INDEX TO ADVERTISERS.

Name.	Manufacturers of or dealers in	Post office address.	Page.
J			
Johns-Manville Co., H. W......	Packings, Bitumen-Mastic Solution, Enamel and Cement, Audiffren-Singrun Refrigerating Machines, Navy Brand Fire Felt, "Noark" Enclosed Fuses, Pipe Coverings — Magnesia, and Asbestos.	296 Madison ave., New York City.	xxxv
K			
Kinney Manufacturing Co.	Kinney High Pressure Pumps............	Boston, Mass..........................	. ..xxxvii
Kelvin & Wilfrid O. White Co.	Adjuster of Ships' Compasses. Contractor for Navigational Equipment.	112 State st., Boston, Mass......	xxxi
L			
Lalor Fuel Oil System Co........	Automatic Stop-Valves.....................	527-529 Colvin st., Baltimore, Md.	xxxiii
Lidgerwood Mfg. Co...............	Hoisting Engines, Electric Hoists, Conveying Machinery.	96 Liberty st., New York.......	xxvi
Lovekin, Luther D..................	The Lovekin Improved Automatic Assistant Cylinder for Valve Gears.	Mill Creek Road, Ardmore, Pa.	xxv
Lovekin Pipe Expanding and Flanging Machine Co.	Pipe Expanding and Flanging Machine.	421 Chestnut st., Philadelphia, Pa.	vi
M			
Mason Regulator Co.............	Mason Reducing Valves and Pressure Regulators.	Boston, Mass..........................	xlii
Metal & Thermit Corporation..	Thermit Quarterly	Equitable Building, 120 Broadway, New York City.	xi
N			
Newport News Shipbuilding and Dry Dock Co.	Ships and Engines..........................	Newport News Shipbuilding and Dry Dock Co., Newport News, Va.	 xviii
Niclausse Boiler Co................	Niclausse Water-Tube Boilers, Niclausse Automatic Stokers.	24 Rue des Ardennes, Paris, France.	 xxiv
P			
Parsons Marine Steam Turbine Co., Ltd.	Marine Steam Turbines.....................	2 Rector st., New York City...	 xiii
Phosphor-Bronze Smelting Co..	Phosphor-Bronze and Delta Metal Castings, Forgings, etc.	2200 Washington ave., Philadelphia, Pa.	xi
Pneumercator Company.......; Pneumercators, F. S. Dudgeon, Ltd.	Pneumercators..................................	15 Park Row, Park Row Building, New York, N. Y............ London, England....................	xl
Power Specialty Co................	Builders of the Famous Foster Superheater. Supply Duval Metallic Packing.	111 Broadway, New York.......	iv
R			
Richardson — Phenix Co.........	Lubrication Engineers and Manufacturers.	114 Reservoir ave., Milwaukee, Wis.	 xliv
Roelker, H. B.........................	Allen Dense-Air Ice Machines, Screw Propellers. Consulting Engineer.	41 Maiden Lane, New York....	iv
Row & Davis, Engineers, Inc ...	Paracoil Feed Water Heaters and Evaporators.	90 West st., New York............	 xxiii
S			
Schmidt Co., F. J.......	Naval and Army Outfitters..................	63 Maryland ave., Annapolis, Md.	xlii
Schutte & Koerting Co............	Fuel Oil Burning Systems, Film Water Heaters, Ship Maneuvering Valves, Ship Emergency Valves, Marine Free Exhaust Valves, Reducing Valves, Distillers, Evaporators.	12th and Thompson sts., Philadelphia, Pa.	 xlv
Seattle Construction and Dry Dock Co.	Shipbuilders, Engineers, Boiler Makers, Founders.	Seattle, Wash..........................	xxxiii
Sterling Engine Company........	Sterling Engines...............................	1264 Niagara st., Buffalo, N. Y.	xxxix

Name.	Manufacturers of or dealers in	Post office address.	Page.
S			
Sugar Apparatus Manufacturing Co., Wilmington, Del. Wheeler Condenser & Engineering Co., Carteret, N. J.	Evaporating Apparatus	Carteret, N. J.	xvi
T			
Terry Steam Turbine Co.	Steam Turbines	Hartford, Conn.	xix
Texas Company, The	Texaco Lubricants, Texaco Ursa Oil, Texaco Cylinder and Machine Oils, Texaco Fuel Oil.	17 Battery Place, New York City.	xiv
U			
United States Metallic Packing Co.	Piston Rod and Valve Stem Packings	427 North 13th st., Philadelphia, Pa.	xxxiii
V			
Vibration Specialty Co.	Vibrations	Harrison Bldg., Philadelphia, Pa.	xii
Volz, William E., M. E.	Positive Patent Lifting Clamp	6 Church St., New York City	xxvii
W			
Ward Engineering Works, The Charles.	Patent Marine Water Tube Boilers	Charleston, W. Va.	[illegible]
Warren Steam Pump Co.	Pumps and Condensing Apparatus for Marine Installations.	Warren, Mass.	xxx
Watson & McDaniel Co.	Steam Traps, Separators, Reducing Valves, etc.	124 N. 7th st., Philadelphia, Pa.	xv
Westinghouse Electric and Manufacturing Co.	Electric Machinery of all classes	East Pittsburgh, Pa.	xxix
Westinghouse Machine Co.,	Marine Reduction Gears	East Pittsburgh, Pa. Washington, D. C.	v
Whitman, Ray B.	Attorney and Engineer	505 Fifth Ave., New York	xlvi
Wheeler, Manufacturing Co., C. H.	Radojet Air Pumps	18th and Lehigh Avenue, Philadelphia, Pa.	xxxvi
Worthington Pump and Machinery Corporation.	Blake Pumps for Naval and Merchant Marine.	115 Broadway, New York City.	vii
Y			
York Manufacturing Co.	Ice Making and Refrigerating Machinery exclusively.	York, Pa.	xli

JOURNAL

OF THE

AMERICAN SOCIETY OF NAVAL ENGINEERS

VOL. XXXI. MAY, 1919. No. 2.

U. S. S. *NEW MEXICO.*

DESCRIPTION AND OFFICIAL TRIALS.

BY COMMANDER S. M. ROBINSON, U. S. N., MEMBER, AND
HENDERSON B. GREGORY, ASSOCIATE.

The electrically-propelled battleship, U. S. S. *New Mexico* is one of three vessels authorized by an Act of Congress approved June 30, 1914. She is officially known as Battleship No. 40, and is a sister ship of the U. S. S. *Mississippi* and U. S. S. *Idaho,* which are equipped with direct-drive Curtis and Parsons turbines, respectively, and geared cruising turbines.

The contract for the electrical propelling machinery for the *New Mexico* was awarded to the General Electric Co., June 3, 1915, at a cost of $431,000.00, and four months later, October 14, 1915, the keel of the vessel was laid at the Navy Yard, Brooklyn.

The *New Mexico* was launched April 23, 1917, and was commissioned on May 20, 1918, since which date she has been subjected to numerous and extensive tests, as noted elsewhere.

PRINCIPAL HULL DIMENSIONS.

Length between perpendiculars, feet and inches	600–0
on L.W.L., feet and inches	600–0
over all, feet and inches	624–0
Breadth, extreme, on L.W.L., feet and inches	97–4½
molded, feet and inches	97–0
Depth molded, main deck at side M.S., feet and inches	46–3
Draught, mean, to L.W.L., feet and inches	30–0
Displacement corresponding, tons	32,000
per inch at L.W.L., tons	100.9
Ratio of length to beam	6.16
Coefficient of fineness, block	0.635
midship section	0.981
L.W.L. plane	0.726

GENERAL DESCRIPTION OF HULL.

In profile the *New Mexico* closely resembles the *Pennsylvania* class. She has one smoke pipe, four turrets and two masts of the customary cage type, which are provided with spotters' tops, wireless, signal yards, etc. There are eight 36-inch searchlights, four on each mast, and four 24-inch signal searchlights mounted on the signal bridge.

Superstructure Deck and Bridges.—The superstructure deck extends from turret II to frame No. 87. On it are mounted four 5-inch guns, the anti-aircraft guns, saluting guns, stowage of boats, foundry and blacksmith shop, deck radio station, etc. Above the superstructure deck is the bridge deck, where the chart house is located. The navigating bridge is on top of the chart house and just behind the conning tower.

Upper Deck.—This deck extends from the bow aft about half the length of the vessel. Forward are the windlass, an electric deck winch and turrets I and II. Aft of the turrets a portion of the 5-inch battery is mounted in a casemate, which also contains the officers' and crew's galleys, butcher shop, bakery, etc.

Main Deck.—The main deck is a weather deck abaft the break in upper deck. In the covered portion forward are

crew's quarters, and aft on the weather portion are turrets III and IV, two electric deck winches and an electric capstan.

Second Deck.—From forward aft are located chief petty officers' quarters, sick bay, crew's space, junior and warrant officers' quarters, wardroom officers' quarters, and the Captain's and Admiral's quarters.

Third Deck.—On this deck, beginning forward, are stores, windlass machinery, prison, firemen's washrooms, coppersmith's outfit, boat crane machinery, radio room, cold-storage rooms, laundry, general workshop, engineer's tool issuing room, crew's space, provisions, and wardroom staterooms aft.

The first and second platforms and the hold are next below in the order named.

COMPLEMENT.

The ship's normal complement, not including additional personnel for flagship, is approximately as follows:

Commanding officer	1
Wardroom officers	26
Junior officers	18
Warrant officers	12
Crew, including chief petty officers	1,011
Marines	72
Total	1,140

BATTERY.

The main battery comprises twelve 14-inch guns, arranged in four three-gun turrets along the center line of the vessel. Turrets I and II are grouped forward on the upper deck, the latter firing over the top of the former, and IV and III are similarly arranged on the main deck aft.

The secondary battery, of fourteen 5-inch rapid-fire guns for torpedo defense, is divided starboard and port as follows: four on superstructure deck and ten within inclosed portion of upper deck.

The 5-inch battery is served by eighteen electric-chain ammunition hoists, each driven by a 3-horsepower motor.

The following smaller guns are also provided:

4 3-inch anti-aircraft guns;
4 6-pounder guns for saluting;
2 1-pounder guns for boats;
2 3-inch field pieces;
2 0.30-caliber machine guns.

The torpedo equipment consists of four 6.8-meter by 21-inch submerged torpedo tubes.

SMALL BOATS CARRIED.

The following small boats constitute the ship's regular allowance:

2 50-foot steamers;
2 35-foot motor boats;
2 50-foot motor sailing launches;
2 40-foot motor sailing launches;
1 40-foot Admiral's barge;
1 31-foot racing cutter;
2 30-foot whale boats;
2 20-foot dinghies;
2 14-foot punts.

All boats are carried on the superstructure deck, port and starboard, except the whale boats, which are hung in davits at the ship's side abreast the after cage mast.

Two electrically-operated boat cranes are provided for handling all boats, except the whale boats. Each crane has two operating gears, one for turning and the other for hoisting, each driven by a 50-horsepower motor.

ANCHOR WINDLASS.

An electric windlass, designed by the American Engineering Co., is located on the upper deck forward, with driving gear below on the first platform. The windlass is driven by

two 125-horsepower motors, through worm gearing and shafting, and so arranged that the wildcats can be operated together or independently of each other.

DECK WINCHES.

Three electric-driven, compound-geared deck winches are provided, one forward on the upper deck, abaft of the windlass, and two aft of the main deck, port and starboard, forward of turret III. They are of the American Engineering Co.'s type, each driven by a 45-horsepower motor.

CAPSTAN.

An American Engineering Co.'s electric capstan is located on the main deck, center line, abaft turret IV, with driving motor below on the third deck. It is compound geared and driven by a 45-horsepower motor.

STEERING GEAR.

The steering gear is of the Waterbury Tool Company's electro-hydraulic type, operated by 75-horsepower motor, fitted in duplicate.

LAUNDRY.

The laundry is located on the third deck, starboard side, midship. It is equipped with the following machinery, all motive power being electricity:

2 Universal presses;
1 Combination ironer;
1 Flatwork ironer;
1 Collar shaper;
2 Extractors;
1 Drying room, 5 drawers;
1 Tumbler dryer;
1 Disinfector;
1 Washer;

1 Ironing table;
1 Soap tank;
1 Starch kettle;
1 2-compartment tub;
3 Truck tubs.

GALLEY OUTFIT.

Officers' Galley.—The officers' galley is equipped with oil-burning range of four sections.

Crew's Galley.—In the crew's galley are installed the following:

1 oil-burning range of eight sections;
4 steam-jacketed copper kettles (80 gallons each);
2 steam-jacketed copper kettles (60 gallons each);
2 coffee urns, 100 gallons each;
1 electrically-driven meat grinder;
1 electrically-driven kitchen and cake machine;
1 hand operated meat slicer;

Bakery.—The following equipment is installed in the bakery:

2 No. 1 U. S. Navy standard, electric bake ovens;
1 steam box;
2 portable dough troughs;
1 electrically-driven dough-mixing machine;
1 electrically-driven kitchen and cake machine;

Miscellaneous equipment located as noted:

1 electrically-driven potato peeler, in potato-peeler room;
1 electrically-driven dish washer, capacity 6,000 pieces per hour, in general mess pantry;
4 warming ovens, in serving room;
1 electric butter slicer, in general mess issuing room.

GARBAGE PLANT.

For disposal of garbage, one oil-burning incinerator is located on the superstructure deck, abreast of the smoke pipe, starboard side.

DRAINAGE SYSTEM.

The four fire and bilge pumps in the center engine room have 5-inch suctions from the after drainage system, which serves compartments aft of number three fire room.

The main circulating pumps, also located in the center engine room, are arranged for pumping on the engine-room bilges in emergencies. There are two bilge connections for the center engine room, and one for each wing engine compartment; each connection is 21 inches diameter.

Each fire room is normally drained by the fire and bilge pump located therein. For extreme emergencies two electrically-driven, vertical, centrifugal pumps are provided for each fire room, with driving motors, 39-horsepower each, located above on the third deck. These pumps have 12-inch suctions and 10-inch discharges.

The drainage of compartments forward, except the forward dynamo room and double-bottoms beneath, is handled by an independent system, served by a horizontal, motor-driven, centrifugal pump. The driving motor is 22.3-horsepower. The latter are drained by the fire and bilge pump in fire room No. 1.

FIRE MAIN.

The fire main is supplied by seven fire and bilge pumps, located in the center engine room and fire rooms.

The main is entirely below the third deck and extends throughout the machinery spaces, close under the deck, in two 8-inch lines, port and starboard. It is cross-connected at the forward and after ends, and in the engine compartments, and supplied by 5-inch risers from the fire and bilge pumps, with cutout valves at the main. From the forward and after cross-connections, 5-inch branches extend in single line forward and aft, reducing in size at the ends 4 inches. Branches to fire plugs are provided as required.

On the main deck forward is a connection to the independent sanitary system for the crew's water closets and washroom. There is a stop valve where it joins the latter.

SANITARY SYSTEM.

The sanitary main is 5 inches in diameter, and is supplied by five 5-inch risers from the fire and bilge pumps, a 5-inch connection from the fire main in the center engine room, and two 5-inch by-passes, one forward and one aft, from the fire main.

Branches, as required, are led to the chief petty officers' washroom and water closets, sick bay, bath, laundry, general mess pantry, galleys, bakery, firemen's washrooms, junior, warrant and wardroom officers' lavatories and water closets, etc.

Forward there is an independent system for the exclusive use of the crew's washroom and water closets. This system is supplied by two 9.3-horsepower, motor-driven, direct-connected, centrifugal pumps, of about 500 gallons per minute capacity each. The main is 5 inches in diameter, with the necessary branches to the plumbing fixtures.

FRESH WATER SYSTEM.

The following fresh-water tanks are provided:

	Gallons.
Ship's tanks	57,910
Mess attendants' washroom supply tank	100
Chief petty officers' washroom supply tank	150
Firemen's washroom supply tank	500
Battle-dressing station supply tank	250
Laundry tank	200
Crew's washroom tanks	450

The ship's tanks have 2½-inch filling connections from the ship's sides, and also a 2½-inch connection from the distiller main. From these tanks water is pumped through the fresh-

water main by means of three direct-connected electric centrifugal pumps of 200 gallons capacity per minute each, driven by motors of 7.75-horsepower each. Branches are led as required from the main to the various lavatories, pantries, etc., and the small gravity tanks enumerated above.

REFRIGERATING PLANT.

The ice-machine room is on the second platform, port side. between the engine and fire rooms. The plant consists of the following apparatus:

2 Kroeschell, vertical, motor-driven, double-acting CO_2 compressors, each capable of producing the ice-melting effect of 6 tons of ice per day. Driving motors, 15-horsepower each.
2 Brine-circulating pumps of the horizontal, motor-driven, centrifugal type. Driving motors, 3-horsepower each.
2 Water-circulating pumps of same type and horsepower as the brine-circulating pumps.
1 Double-pipe, horizontal, CO_2 condenser.
1 Brine-cooling and storage tank.
2 9-can ice boxes.
1 Ice cream freezer.
1 400-quart hardening box for ice cream.
2 Scuttle-butt coils.
4 Cold-storage rooms.

The cold-storage compartment is located on the third deck, immediately above the ice-machine room, and consists of four rooms, one each for meat, butter, officers, and the crew, the latter serving also as vestibule. The rooms are heavily insulated and fitted with cooling coils of 1¼-inch galvanized iron pipe. The following data is given for the cold-storage compartments:

Compartment.	Volume, cu. ft.	Cooling surface of coils, sq. ft.	Temperature to be maintained.
Meat room	2,100	960	15°F.
Butter room	152	90	40°F.
Crew's room	422	200	40°F.
Officers' room	460	230	40°F.

Except the ice boxes, which are direct cooled by CO_2 coils submerged in a brine bath surrounding the ice cans, the plant is operated on the brine-circulating system. The brine is cooled in the cooling tank, also provided with CO_2 coils, submerged in the brine, and then circulated through the system by the brine-circulating pumps. The return brine is led back to the cooling tank, where it is again cooled and the cycle repeated.

RESERVE FEED TANKS AND CONNECTIONS.

Double-bottom compartments B-1 to B-4, inclusive, are fitted up as reserve feed tanks.

The tanks are filled through a 4-inch pipe fitted with two 2½-inch hose connections at the ship's sides, port and starboard, and connected to the combined filling and suction manifold in fireroom No. 2.

There is also a 2½-inch filling pipe from the distiller fresh-water main, which is led to the same manifold. From the manifold 4-inch combined filling and suction pipes are led to the bottom of each tank.

There is a 4-inch main between the manifold and the auxiliary-feed pump suctions, so that the auxiliary-feed pumps may draw from any tank. The auxiliary-feed pump in fireroom No. 2 is arranged to draw direct from any tank and discharge to any other tank. The reserve feed pump in the engine-room also has a 4-inch suction from the manifold, for drawing water direct from any tank and discharging same to the main feed tanks.

VENTILATING SYSTEM.

Artificial ventilation is provided where necessary for all quarters, living spaces, passages, storerooms, magazines, engine rooms, dynamo rooms, evaporator room, etc.

In general air is supplied on the plenum system to the different compartments requiring ventilation. The toilet spaces are also provided with the exhaust system.

The ventilating fans are of the B. F. Sturtevant Company's type, driven by G. E. Co. motors, except for a few minor exceptions.

Heater boxes are fitted in the ventilating ducts to all quarters below the main deck, crew's space, etc., for heating the incoming air in cold weather; no other heating apparatus being provided for these spaces.

HEATING SYSTEM.

All staterooms and quarters (except the captain's, admiral's and chief-of-staff's quarters), crew's space, etc., are heated by the thermo-tank ventilating system; heater boxes provided with suitable steam coils are placed in the air duct for heating the air supplied to these compartments. The heater boxes are of the Zimmerman type, as manufactured by the Schutte-Koerting Co. The remaining portions of the vessel, except the turrets, torpedo room and charthouse, which are provided with electric heaters, are heated by the customary pipe-coil steam radiators.

The heating system is divided into two main sections, one forward and one aft. The former takes steam from the auxiliary steam pipe in fireroom No. 1, and the latter from the auxiliary steam pipe in the center engine room. Stop and reducing valves are fitted at the branches from the auxiliary steam pipes.

The galley steam is taken off the auxiliary steam line in firerooms Nos. 2 and 3; a stop valve and a reducing valve are provided at each connection.

The drains from the heating system and galley are led to suitable traps, which discharge to the feed and filter tanks and condensers.

MAIN PROPELLING MACHINERY AND MAIN AND EXCITER SWITCHBOARDS.

The main propelling machinery of the *New Mexico* consists of two alternating-current turbo-generators, four induction motors, two 300 kw. direct-current generators for excitation and motor-driven auxiliaries, two motor generator boosters, a main switchboard, an exciter switchboard, ventilating blowers for the main motors, and the necessary wire and cable for connecting up the generators and motors to the switchboards.

The weight of the propelling machinery, exclusive of spare parts, is about 500 tons. This is about 215 tons less than the weight of the *Mississippi*, a sister ship of the *New Mexico*.

A section of the main turbine is shown in Figure 1. It is a 10-stage, G. E.-Curtis turbine. The first stage is velocity compounded, having two rows of moving buckets, and the other stages are pressure compounded, having a single row of moving buckets. The wheels are pressed on to the shaft and keyed. The tenth-stage wheel secures against a shoulder on the shaft and the first-stage wheel is secured by a cross key in the wheel and shaft, thus preventing fore and aft motion of the wheels. The wheels are separated approximately .01 inch by crushing pieces inserted between the wheels. The blades are secured to the wheels by dovetails, as shown in Figure 1. All stages except the first have complete peripheral admission. The admission nozzles are cast into diaphragms which separate the successive stages. These diaphragms are split through the horizontal plane and are secured in the casing against shoulders. The lower half of each diaphragm may be rolled out without lifting the rotors. When the casing is lifted, the top half of the diaphragm may be lowered by backing off a screw on each side which takes the

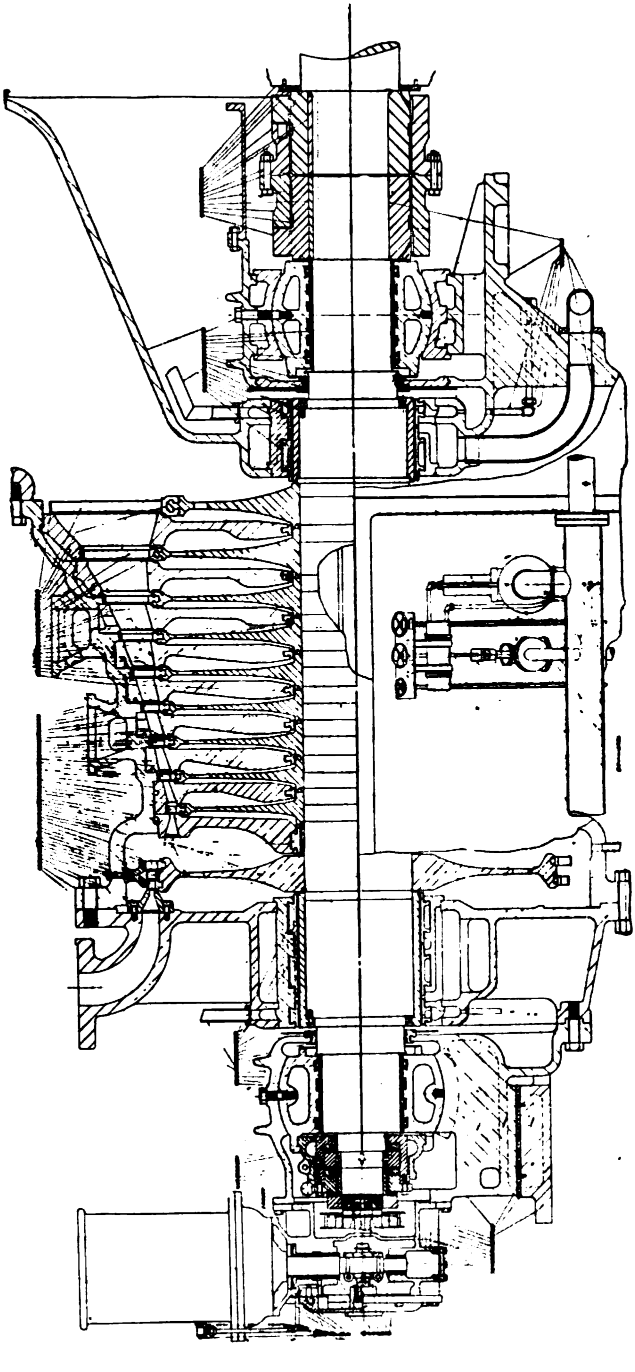


FIG. 1.—CROSS SECTION OF MAIN TURBINE.

weight of the diaphragm when it is lifted from the lower half. Leakage past diaphragms, along the shaft, is prevented by packing rings of the labyrinth type, made of soft brass. The intermediate segment, carrying stationary blading, in the first stage of the turbine is secured by conical-headed bolts through the turbine casing, thus preventing any danger of these coming loose and getting into the moving parts of the machinery. The first-stage expanding nozzles are secured by countersunk, cheese-headed screws; sufficient metal is peened over the heads of these to prevent them from getting into the machinery in case of breakage. The turbine casing is split in a horizontal plane and also in a vertical plane at both high-pressure and low-pressure ends. This makes it possible to lift off the portion of the turbine casing covering the blading without disturbing the exhaust trunk of the turbine and also without removing the steam chest at the high-pressure end of the turbine. The shaft openings in the casing at each end are sealed by labyrinth packing rings which are supplied with sealing steam from the turbine itself when the turbine is loaded sufficiently to give high steam pressure for this purpose; when the pressure falls too low, sealing steam is supplied from the main steam line through a reducing valve. There is an unloading valve on the steam-sealing line between the high-pressure end and low-pressure end which maintains the proper pressure on the sealing line by spilling the excess steam into the eighth stage of the main turbine. The turbine casing is drained through valves, by a drain pump located in the center engine room. The turbine is coupled to its generator by a flexible coupling which is totally enclosed by a housing. The turbine is provided with a bearing at each end mounted on brackets which are integral with the turbine casings. The bearings, which are of the self-aligning type, are supplied with forced lubrication from pumps located in the center engine room. These pumps also supply oil pressure for operating the main gover-

nor hydraulic relay. The bearing at the high-pressure end carries the turbine thrust bearing, which is of the plain collar type. The outside shell of the thrust bearing is provided with worm and worm wheel attachment for moving the turbine in the fore-and-aft direction, to give proper clearance between the buckets and nozzles. There is a clearance indicator at the end of the turbine shaft.

The turbine casing has two openings, one into the fifth and one into the eighth stage, for the admission of exhaust steam, either from the exciting units or from the exhaust line of the ship. These openings are provided with stop check valves and also automatic trip valves. The exhaust line is also provided with a valve for by-passing exhaust steam to the condenser when desired. This valve can be locked open by hand so that the exhaust steam passes direct to the main condenser; it is also spring-loaded and set at ten pounds so that pressure in excess of this will relieve itself into the main condenser; it is also directly operated by the main turbine governor so that when the main turbine governor closes all valves on the steam chest, further motion of it will open this by-pass valve, thus admitting the exhaust steam direct to the condenser and by-passing the main turbine. The stop valves are provided so that steam can be admitted to either stage as desired, and the check feature is added to prevent steam by-passing from the fifth stage to the eighth stage in case both valves should be opened and the steam in the fifth stage of a higher pressure than that corresponding to the auxiliary exhaust pressure. The trip valves are operated by the emergency governor.

The emergency governor is a weight secured to the main shaft and held in place by a spring; overspeeding the turbine will force this weight out against the pressure of the spring, tripping the trip valves of the fifth and eighth stages of the turbine and also tripping the main throttle valve, thus shutting off all sources of supply of steam to the turbine. These

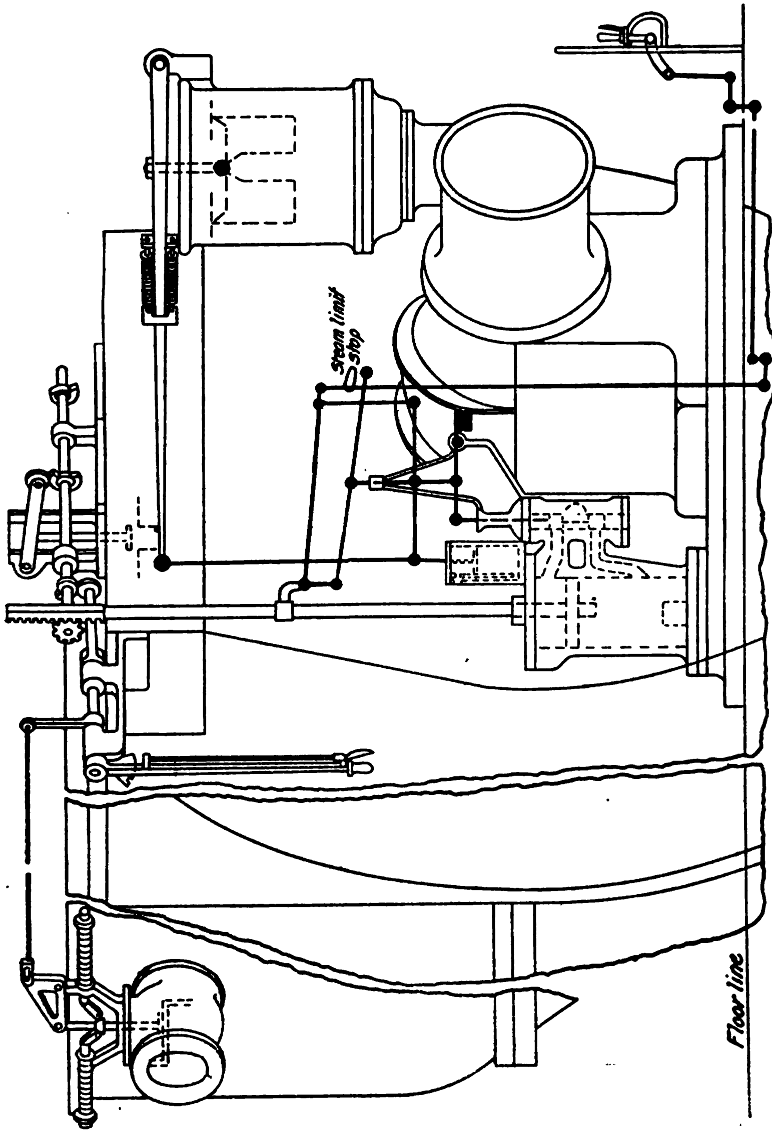
valves can also be tripped by a hand pull located in the center engine room.

The main governor is driven by a worm gear from the high-pressure end of the turbine shaft. It is of the flyball type, the weights being mounted on knife edges and working against a spring. This governor is adjustable and can be set to give any desired speed from 700 revolutions to 2,200 revolutions, and when once set will maintain that speed regardless of changes in load.

A diagrammatic sketch of the governor and its control gear is shown in Figure 2. The governor weights operate a pilot valve of a hydraulic relay, which, in turn, admits oil pressure to a piston driving a rack; this, in turn, drives a pinion which is mounted on a camshaft; motion of these cams opens successive valves in the steam chest of the turbine, thus admitting the desired amount of steam. This control mechanism for the governor is moved from the center engine room by means of rods and bell cranks, as indicated in Figure 2. The effect of the motion of this transmission system is to shift the fulcrum about which the governor weights act and thus to give a different speed for each setting of the fulcrum. In addition to the system of bell cranks for setting the fulcrum, there is another system of bell cranks and rods coming from the center engine room which operates a stop shown diagrammatically in Figure 2. The effect of this stop is to limit the motion of the governor for any given setting, and thus limit the amount of steam that the turbine can take for that setting. This steam limit is necessary in connection with the maneuvering of the ship, as will be explained later on in this article.

A cross section of the main generator is shown in Figure 3, and the assembled rotor is shown in Figure 4. The generator is normally rated at 11,500 kw. at 80 per cent. power factor, and has an overload rating of 25 per cent. It is quarter-phase, two-pole, and designed to run at about 2,100 r.p.m.

The stator windings of the two phases are arranged as



U.S.S. NEW MEXICO.
ARRANGEMENT OF CONNECTIONS OF OPERATING MECHANISM

Fig. 2.

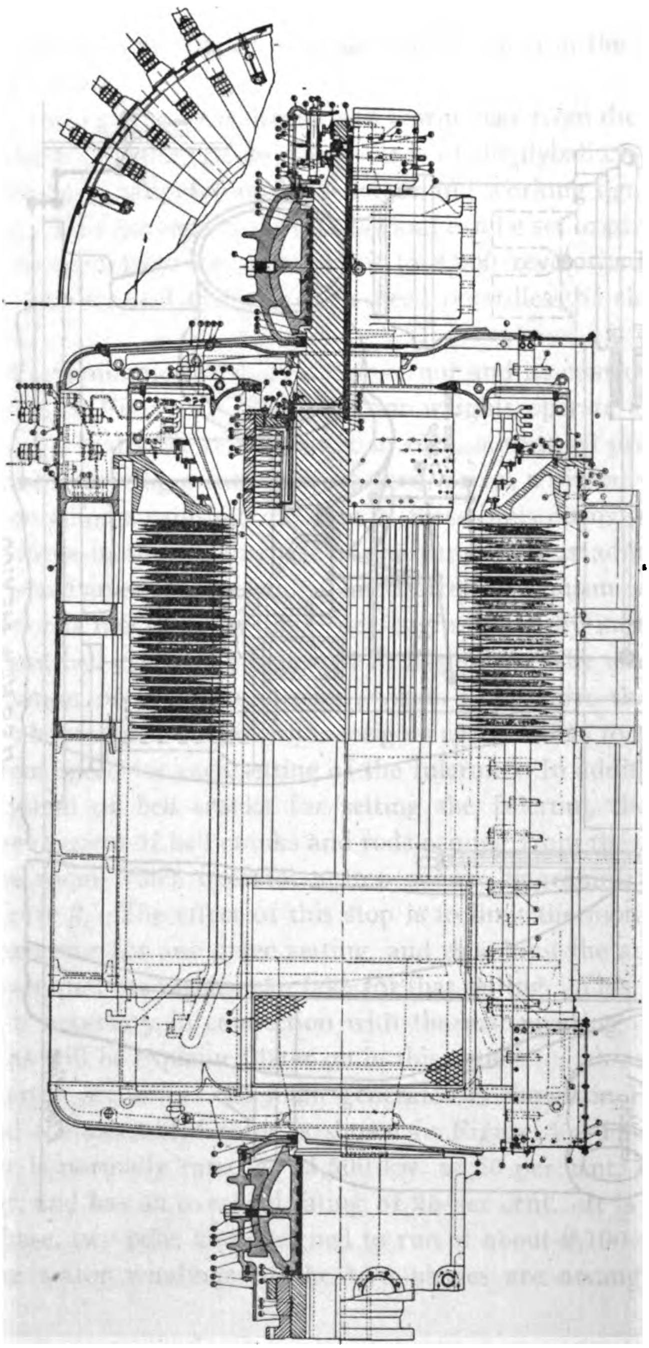


Fig. 3.—Cross Section of Main Generator.

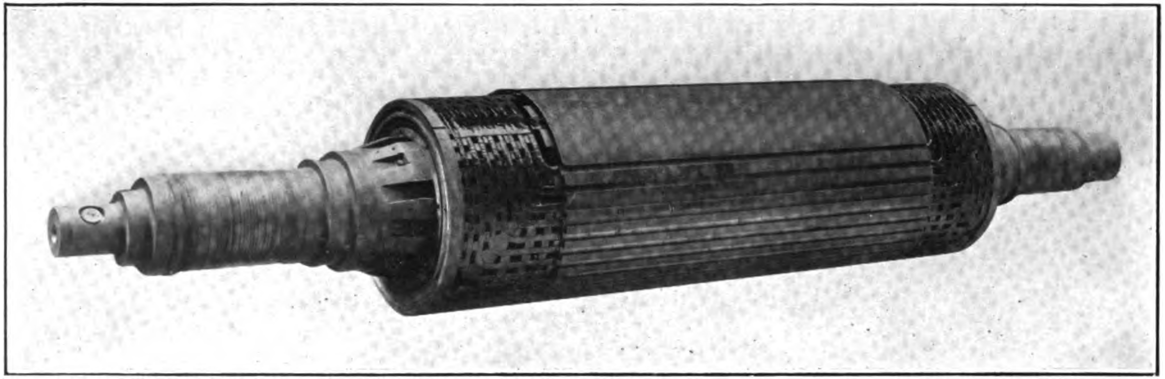


Fig. 4.—Rotor for Main Generator.

shown in Figure 5. Both ends of each independent winding are brought to the terminals of a double-throw switch; this switch, when in one throw, puts the winding in the diametral or parallel connection, and with the generator operating at designed speed and field strength, gives 3,000 volts; with the switch in the other throw, the windings are

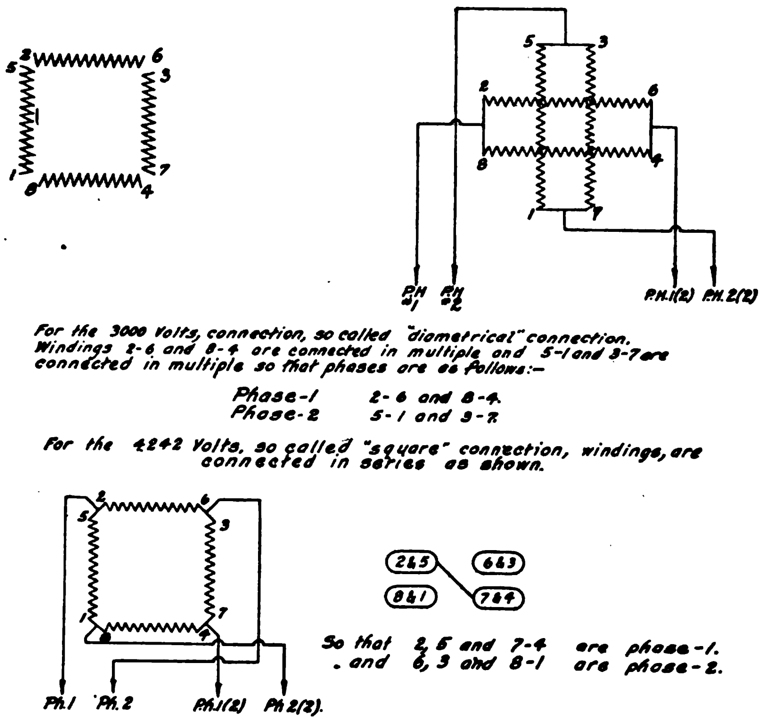


FIG. 5.—MAIN GENERATOR CONNECTIONS.

thrown into the square or series connection and this arrangement at designed field and speed of the generator gives 4,242 volts. The windings are arranged in this way so as to give the maximum efficiency of the generator when one generator is driving four motors and also when it is driving only two motors. In the first case the low-voltage connection is useo and this utilizes the increased current path when using four

motors. When using the high-voltage connection, only two motors are driven from one generator, thus cutting the current path down to one-half its value in the other case. The switch for making this change in "set-up" is mounted on the outboard bulkhead of the center engine room and is operated in the center engine room, and is called the generator disconnecting switch.

The generator rotors are designed with a large margin in regard to heating in order to take care of the condition that exists in backing when over-excitation is applied to the main field. The slip rings for admitting exciting current to the field are external to the bearings, thus reducing the length of the rotor. The generator rotor has fans attached to each end of it for supplying the necessary air for cooling the generator; the outlet for this air is through an exhaust duct to the main deck of the ship. The rotor is a solid steel forging having radial slots machined in it to secure the rotor windings. The rotor windings are insulated by mica and asbestos.

The insulation of the main generator stator consists mainly of mica with fibrous material on the end windings. The entire insulation of the coils is treated to make it moisture proof. Generator stator windings are provided with thermo-couples for measuring rise in temperature; the leads from these thermo-couples are taken to the main switchboard in the center engine room.

The generator bearings are of similar design to the turbine bearings. They are supported on pedestals which are mounted on foundations built in the ship and are not attached to the generator casing.

Excitation for the generator is provided by the 300-kw. generator sets located in the center engine room and by motor-generator boosters. The field current comes from the 300-kw. generator and passes in series through the armature of the generator of the booster. Therefore, if the voltage of the booster generator is varied, the voltage imposed on the slip

rings of the generator field will also be varied. The booster generator is arranged so that its voltage can be either added to or subtracted from that of the 300-kw. generator, thus giving the desired range of field control of the main generator without changing the voltage of the 300-kw. generators. The field of the main generator is opened and closed by the main field switch located on the exciter switchboard in the center engine room.

The 300-kw. generators for supplying excitation and power for the motor-driven engine room auxiliaries are similar to the ship's power and lighting sets, but the turbines are single-stage instead of three-stage, and are designed to exhaust against a back-pressure; they normally exhaust into the main turbine casing as previously explained in this article. The power required for excitation is about 55 kw., so this leaves about 245 kw. to be used for the motor-driven engine-room auxiliaries, which are the main circulating pumps, main air pumps, main condensate pumps, forced-lubrication pumps, and motor-driven ventilating blowers for main motors.

The details of the main motors are shown in Figures 6, 7, 8, 9, 10, and 11. The motors differ more than any other part of the machinery from ordinary commercial practice. It is believed that these are the first motors of this type to be built. They are rated at 7,250-H.P. each, at 167 revolutions, which corresponds to a speed of 21 knots of the ship.

The motor-stator windings are arranged to provide for pole-changing, and there are pole-changing switches located in the center engine room. One throw of these switches will give 24 poles on the stator of the motor and the other throw of the switch will give 36 poles on the stator of the motor. This gives in the first case a 12 to 1 reduction of turbine speed on the propeller, and in the second case gives an 18 to 1 reduction between turbine speed and propeller speed. This enables the turbine to be operated at full speed, both at 21 knots and at 15 knots, and adds largely to the economy of the turbine.

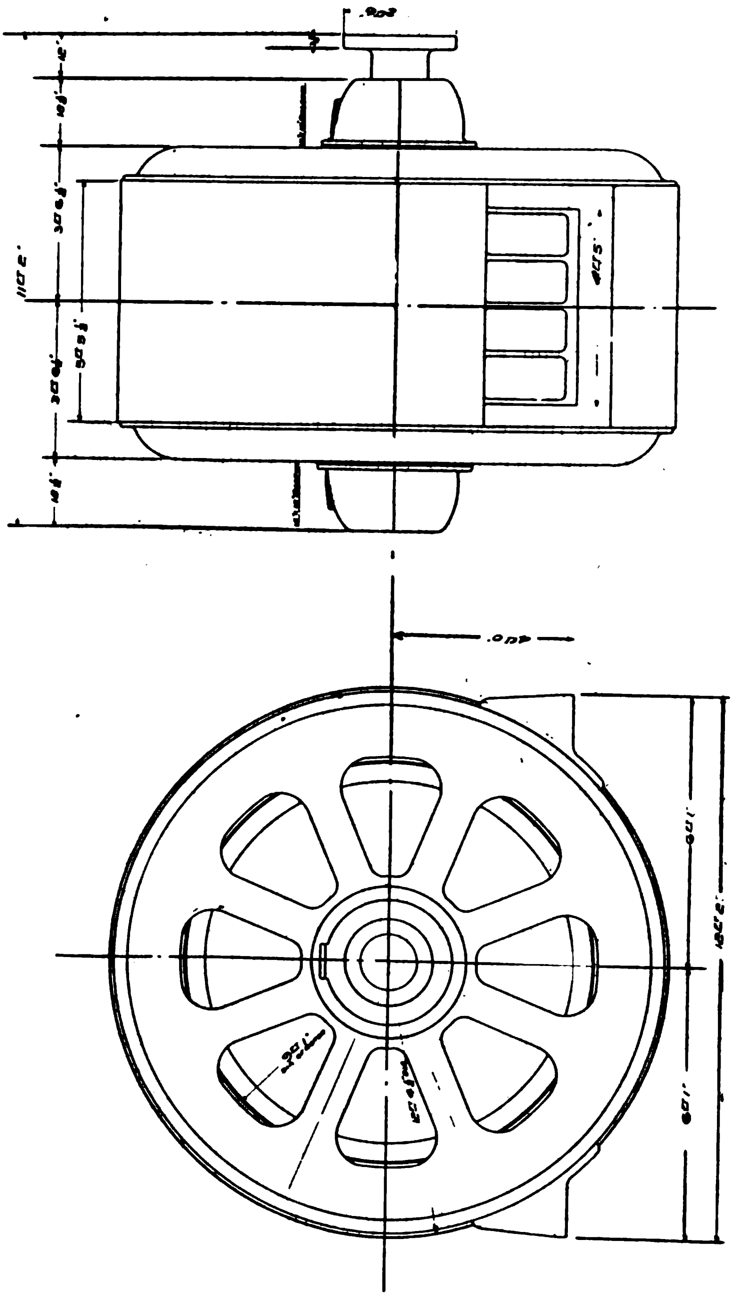


FIG. 6.—OUTLINE OF MAIN MOTOR.

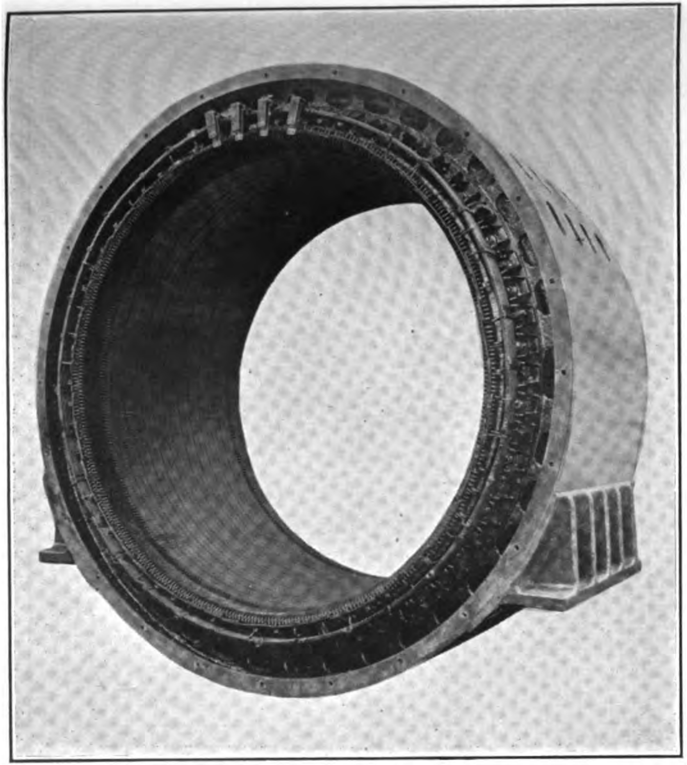


FIG. 7.—STATOR FOR MAIN MOTOR.

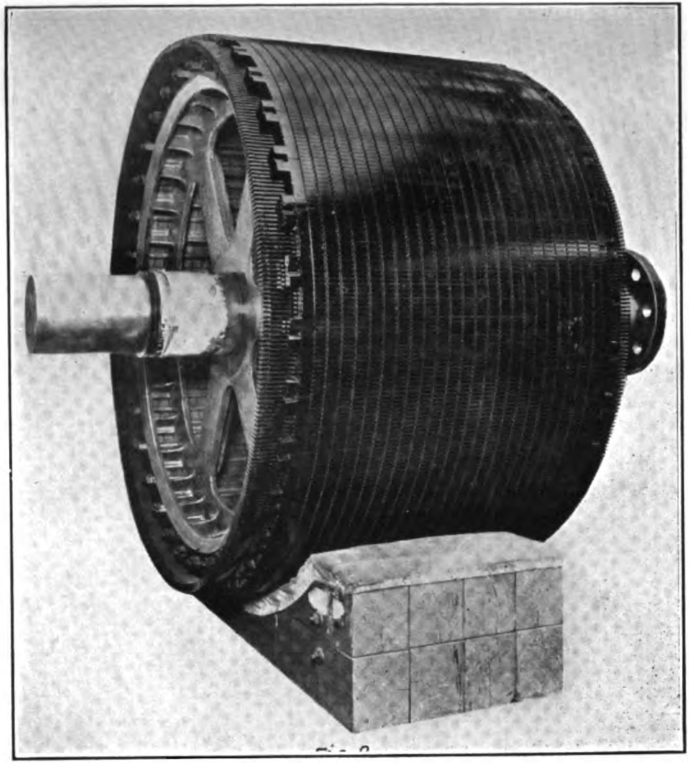


FIG. 8.—ROTOR FOR MAIN MOTOR.

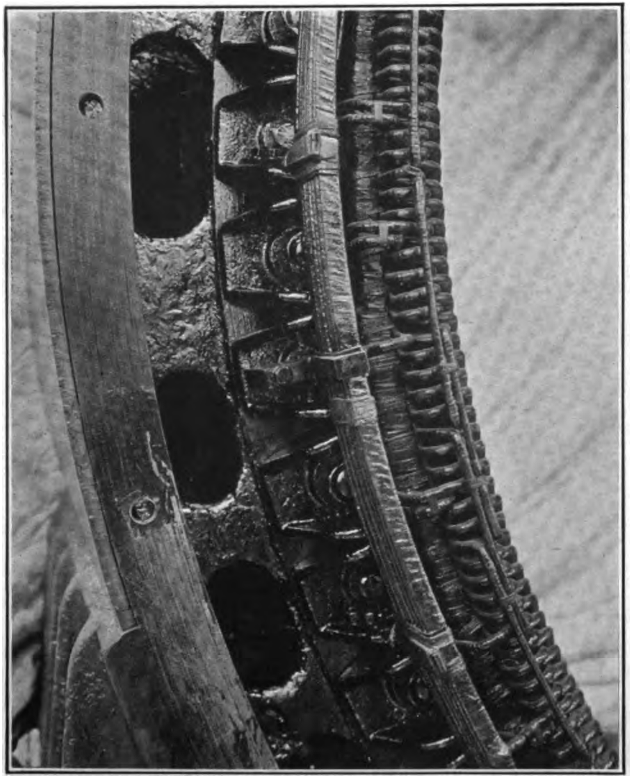


FIG. 9.—STATOR WINDING OF MAIN MOTOR.

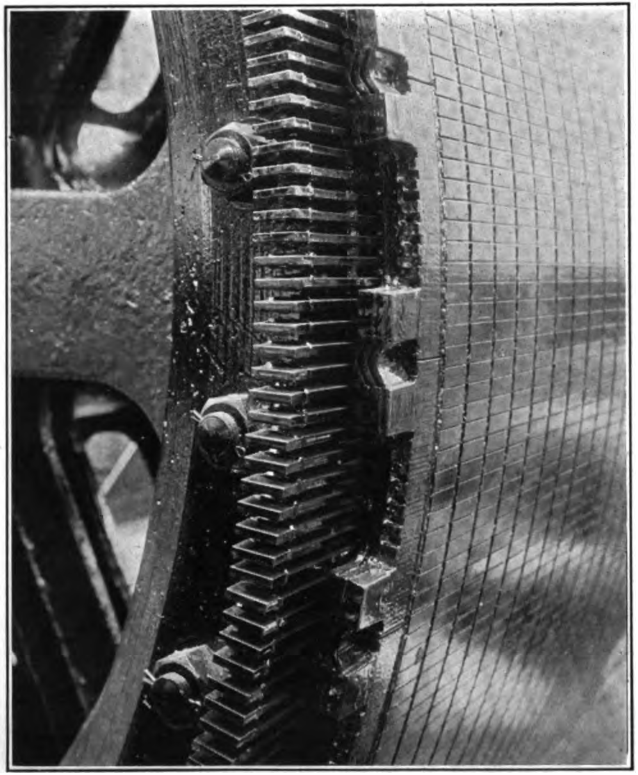


FIG. 10.—ROTOR WINDING OF MAIN MOTOR.

The stator and stator windings are shown in Figures 7 and 9. The insulation is similar to that of the generator stator, consisting mainly of mica, except for the end windings, which are provided with fibrous insulation, the whole being treated to make it moisture proof.

The motor rotor is shown in Figures 8 and 10. The rotor winding is a double squirrel-cage winding. Each slot in the rotor consists of one slot near the periphery of the rotor and one deeper in the iron of the rotor, the two being connected by a long, narrow air gap, as shown in Figure 11. Each of

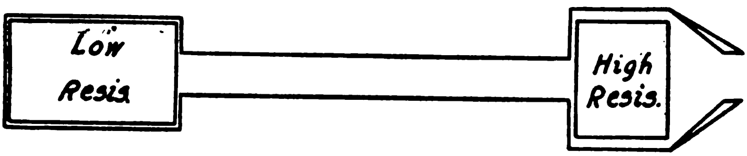


FIG. 11.—ROTOR SLOT.

these slots is provided with an indpendent squirrel cage. The squirrel cage in the outer slot is composed of high-resistance bars and that of the inner slot is composed of low-resistance bars. The high-resistance bars are made of German silver and the low-resistance bars of copper. The bars are wedged tightly into the slots in both cases without insulation. Each squirrel cage has the end of its bars connected by short-circuiting rings; in the case of the high-resistance bars this short-circuiting ring is divided into segments connected by flexible copper connections to provide for expansion, as shown in Figure 10. The object of this construction of the rotor is to provide a winding which will give the necessary torque required for reversing and at the same time make it possible to use the squirrel-cage motor, which is a very simple type of motor with no insulation in its rotor and requiring no changes in its rotor winding when shifting from 36 poles to 24 poles, and vice versa, in the motor stator.

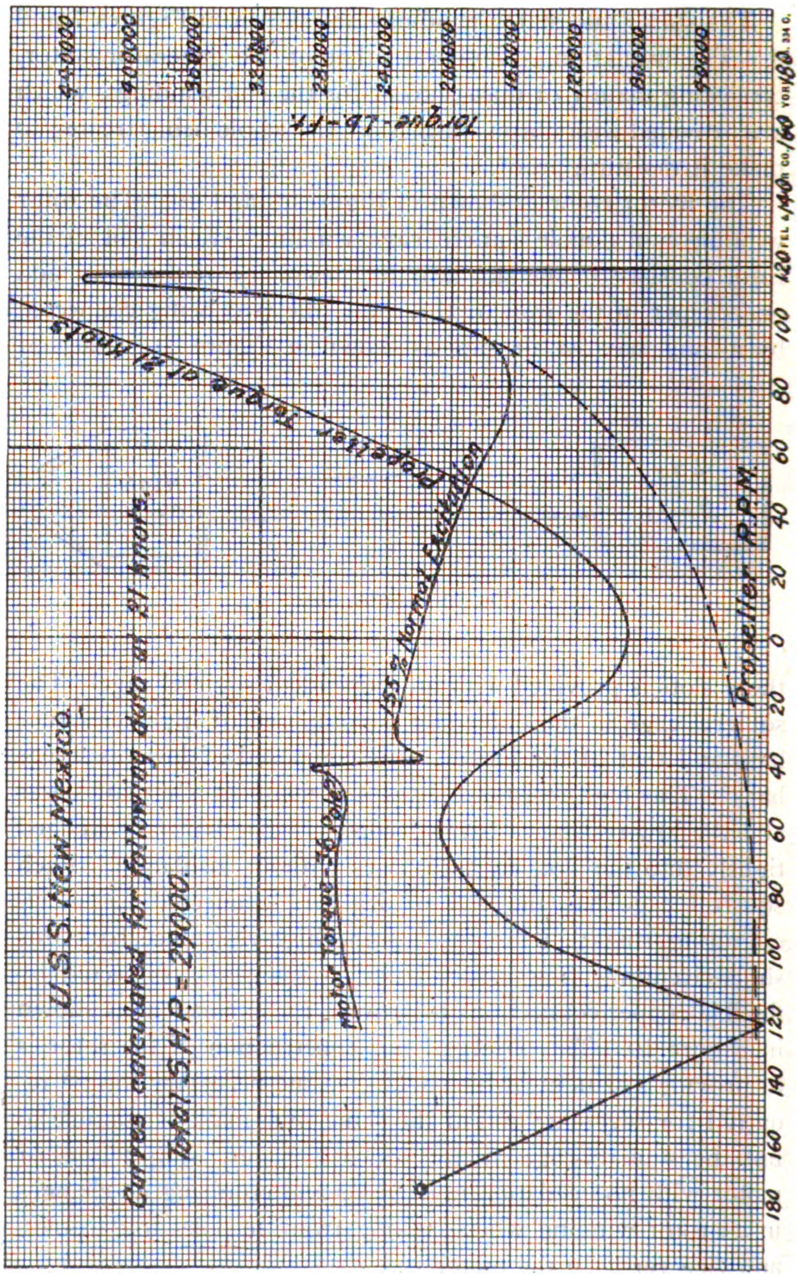


Fig. 12.—Motor and Propeller Torque Curves.

To better understand the requirements of the induction motor for backing, it will be necessary to refer to Figure 12, which gives the shape of both propeller-torque curve and motor-torque curve. It will be seen that the maximum torque required during the process of bringing a propeller to rest, with the ship going ahead, is almost as great as that required to drive the ship ahead at full speed. This is a very severe condition for induction motors to meet, as the torque of an induction motor when "out of step" or out of synchronism with its generator is very small indeed. The dotted portion of the motor-torque curve shows the torque of an ordinary induction motor when out of synchronism with the generator. This is, of course, a motor with low resistance in the rotor. If the resistance in the rotor is increased, the dotted portion of the torque curve may be brought up to that indicated by the heavy portion of the motor-torque curve and this is greater than the propeller-torque curve and will therefore reverse the propeller. It will be seen that the motor and propeller curves intersect each other in the given Figure at about 48 revolutions, or while the motor is still out of synchronism with the generator. It would be impossible to further increase the speed of the motor under this condition, as its torque begins to fall off with increase of speed, but by slowing down the turbine this point of intersection can be moved along on the propeller-torque curve until the propeller-torque curve intersects the motor-torque curve in that portion of the curve to the right of the maximum point, or in the "in synchronism" part of the curve. After the motor has come into synchronism with its generator the two can then be brought up together to whatever speed is desired. The increase in the resistance of the rotor which brings about the increase in the torque of the motor in the "out of synchronism" part of its curve may be best understood by reference to Figure 13. This diagram is an equivalent circuit to that of the double squirrel-cage induction motor. It will be noted that part of the reactance of the two

squirrel cages is mutual and part of it pertains only to the lower squirrel cage. At low frequencies both of these reactances are very small and have no great effect on the current flow, but at high frequencies they become very large and that of the inner squirrel cage becomes very much larger than the outer, thus causing the greater part of the current to flow

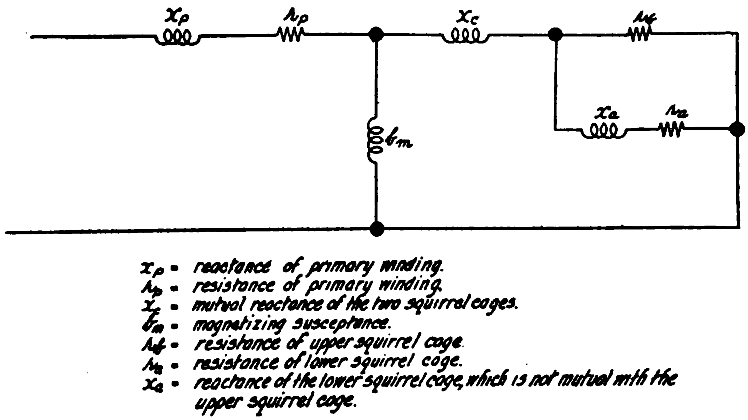


FIG. 13.—EQUIVALENT DIAGRAM OF DOUBLE SQUIRREL-CAGE MOTOR.

through the outer or high-resistance squirrel cage and giving the equivalent of a high-resistance squirrel-cage motor. As the frequency falls, this reactance decreases in value until at the low frequencies in the rotor, such as obtain when going ahead, the reactance virtually disappears and we have the equivalent of an efficient low-resistance squirrel-cage motor. This winding, therefore, gives automatically the equivalent of a high-resistance squirrel cage for backing and a low-resistance squirrel-cage for driving under normal conditions. The physical explanation of what takes place when the motor-stator windings are reversed, thus giving double generator frequency in the rotor, is that the flux in the rotor which normally encircles both squirrel cages is forced across the narrow air gap, practically short-circuiting the low-resistance bar and leaving it out of action almost entirely; as the frequency

falls, due to the reversal of the motor and its coming up to synchronism, more of this flux encircles the lower bar until the low frequencies are reached, when practically all of it again passes around the lower bar. The reversal of the

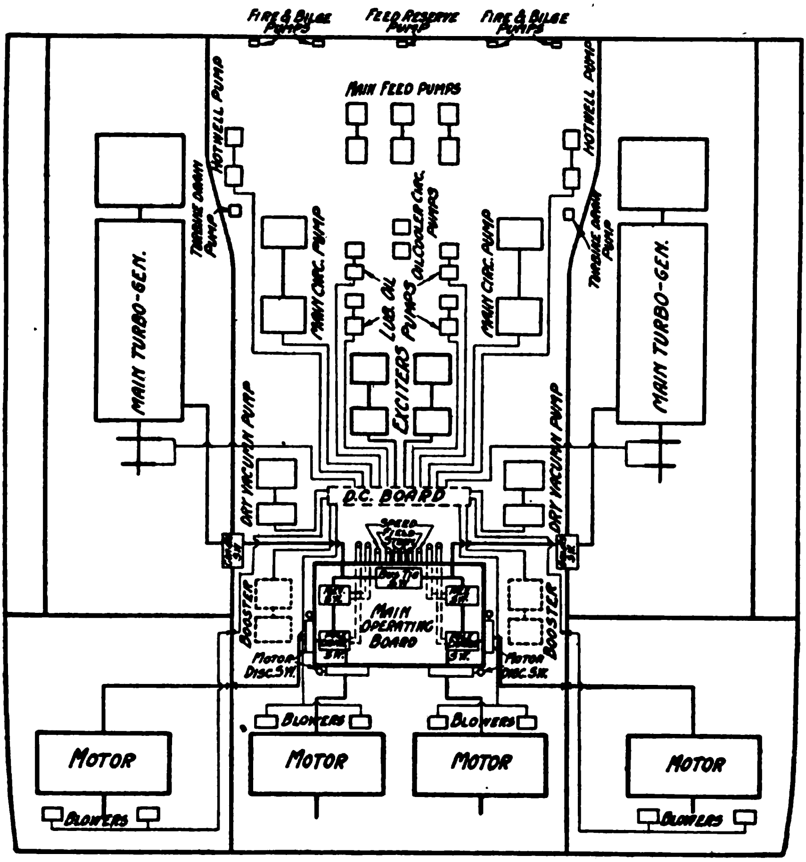


FIG. 14.—ARRANGEMENT OF MACHINERY.

motors is accomplished simply by crossing two of the leads of one phase of the motor, and this is accomplished by reversing switches located in the center engine room.

The main motors are ventilated by suction blowers located one on each side of each motor on top of its casing; these

draw air through the motor and discharge into ducts which are carried up to the atmosphere on the main deck.

The general arrangement of the machinery in the three engine rooms and the motor rooms is shown in Figure 14, which also gives a single line wiring diagram of the main A.C. and D.C wiring. The exciter switchboard is shown in Figure 15. A diagram of the wiring is given in Figure 16. From this wiring plan it will be seen that there are two buses for supplying the auxiliaries and excitation. Either of these buses can be supplied by either of the exciters and also by the ship's circuit from the after distribution board. This makes it possible to always have two live buses in the engine room for the engine-room auxiliaries and excitation. The switch supplying each of these units is made double throw, so that if anything goes wrong with one circuit the auxiliaries may be quickly transferred to the other bus without stopping the operation. This arrangement of the board also makes it possible to use the 300-kw. units in the engine room for exciting the main generator field direct in case of trouble with the booster. In this case the power for driving engine-room auxiliaries on the side of the ship where the booster is disabled would have to be taken from the ship's supply circuit in the after distribution room. Provision is also made for supplying the ship's circuit from either of the exciters; when cruising at slow speeds there will not be much direct current used, and it may be possible to cut out the ship's generators and run only on the two exciting units. This should give very economical operation, as there would be only one condenser in the ship operating and all exhaust steam would be utilized in an efficient manner, either in the main turbine or feed heaters, or evaporators.

The exciter switchboard, in addition to the other direct-current switches mounted on it, has also the main field switch, which is a solenoid-operated switch, and is controlled from the main switchboard. This switch is provided with magnetic blow-out to reduce the arcing on opening the main field,

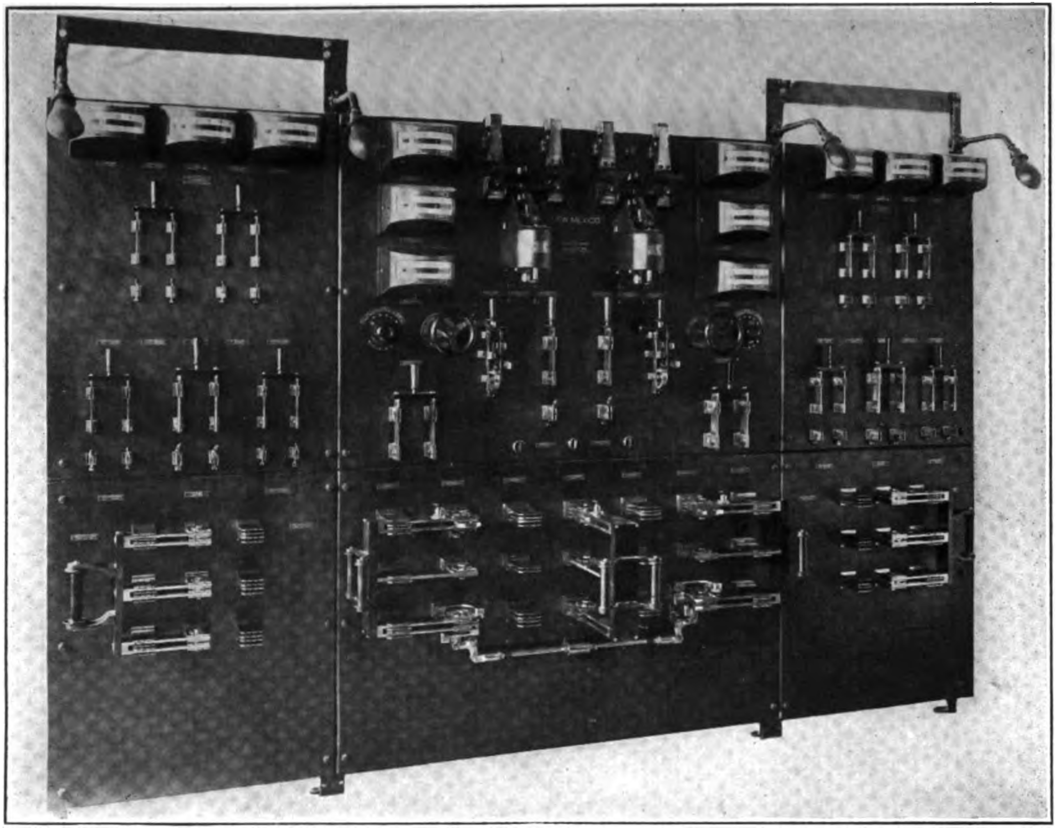
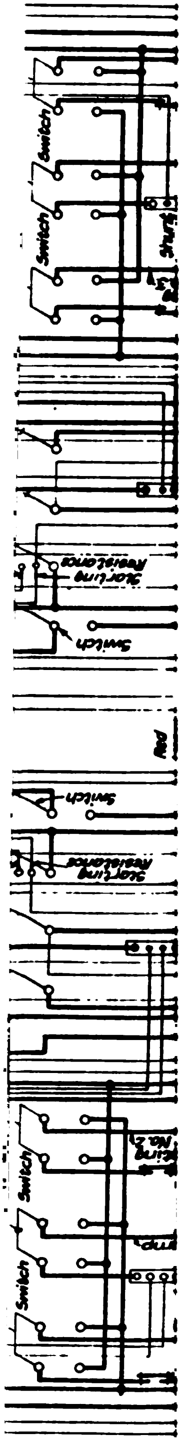


FIG. 15.—EXCITER SWITCHBOARD.


Switch
Switch
Shunt
Switch
Starting Resistance
Red
Switch
Starting Resistance
Switch
Switch

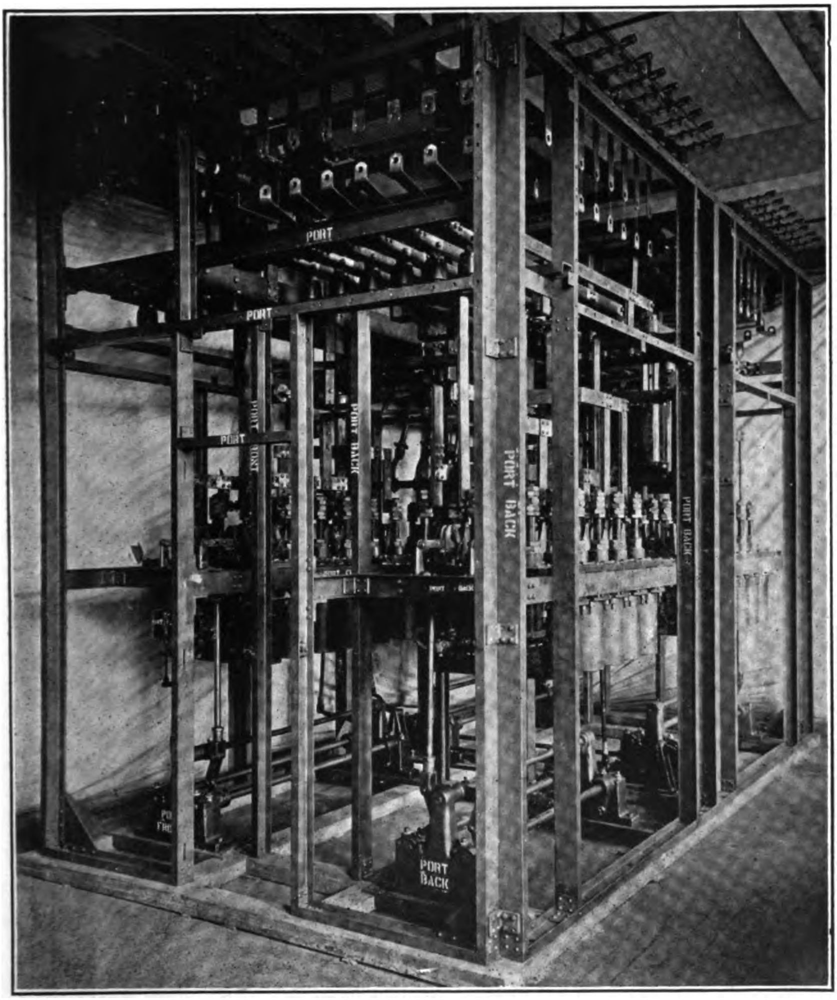


Fig. 17.—Main Switchboard—Back View with Grille Work Removed.

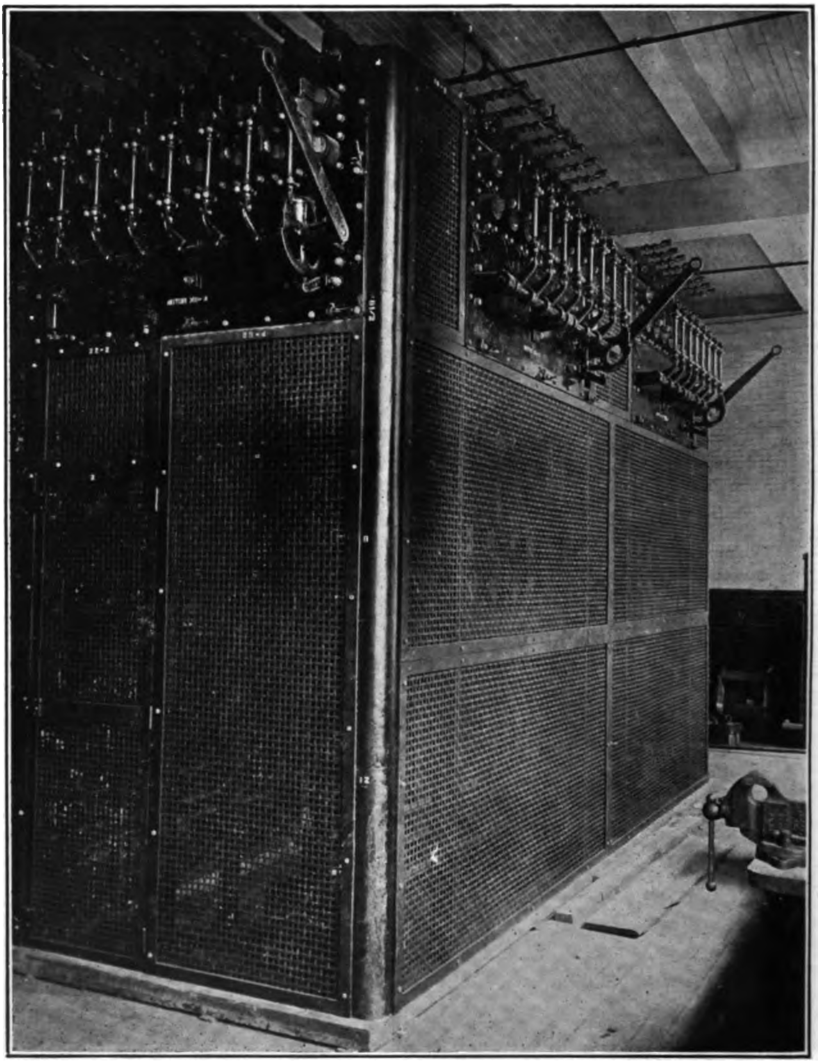


Fig. 18.—Main Switchboard—Back View.

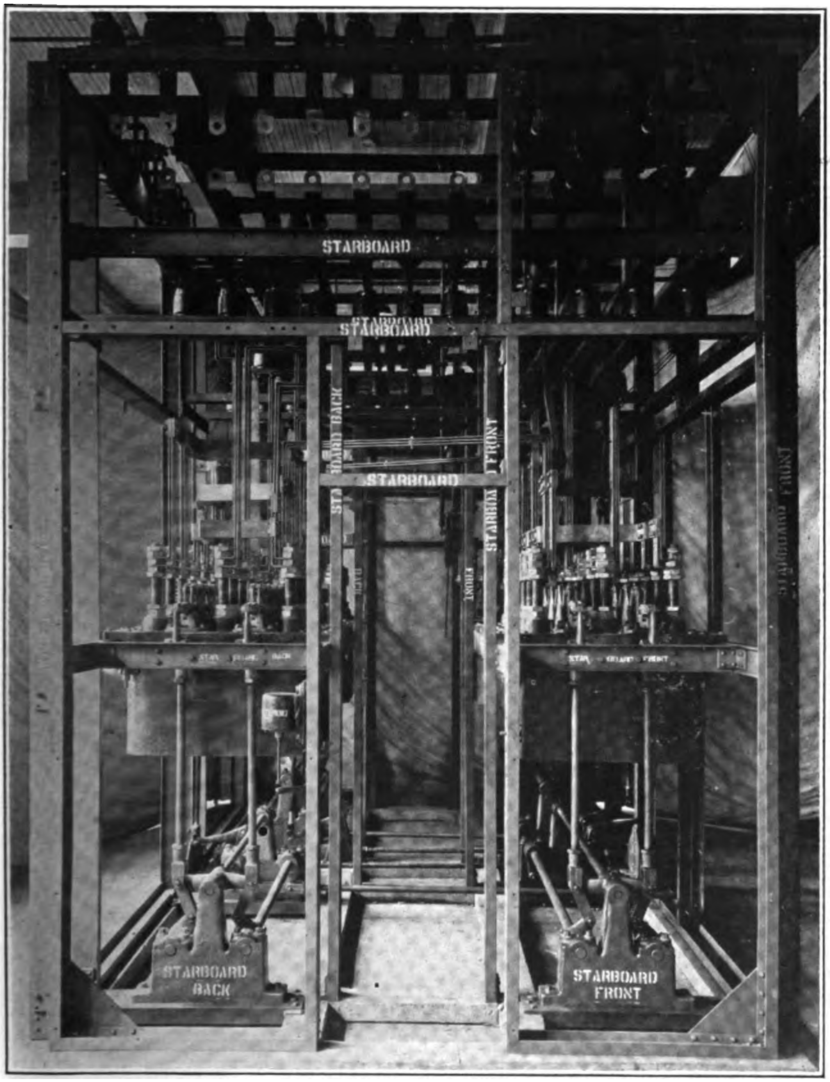


FIG. 19.—MAIN SWITCHBOARD—SIDE VIEW WITH GRILLE WORK REMOVED.

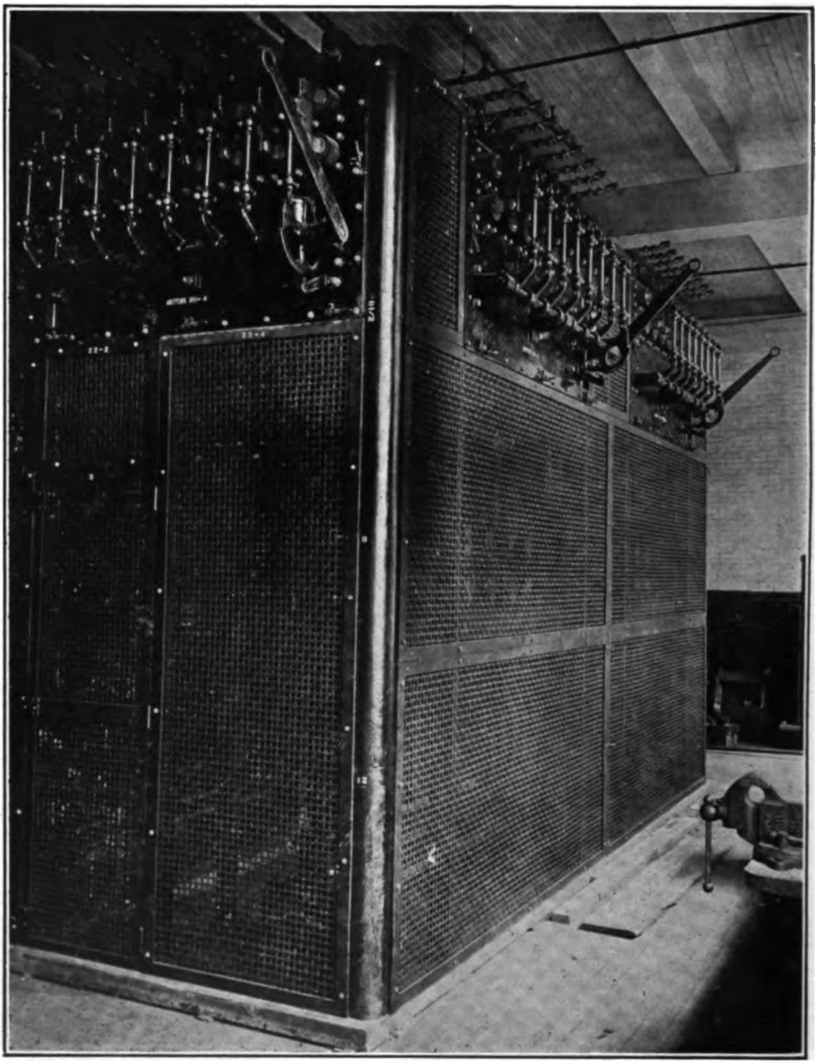


FIG. 18.—MAIN SWITCHBOARD—BACK VIEW.

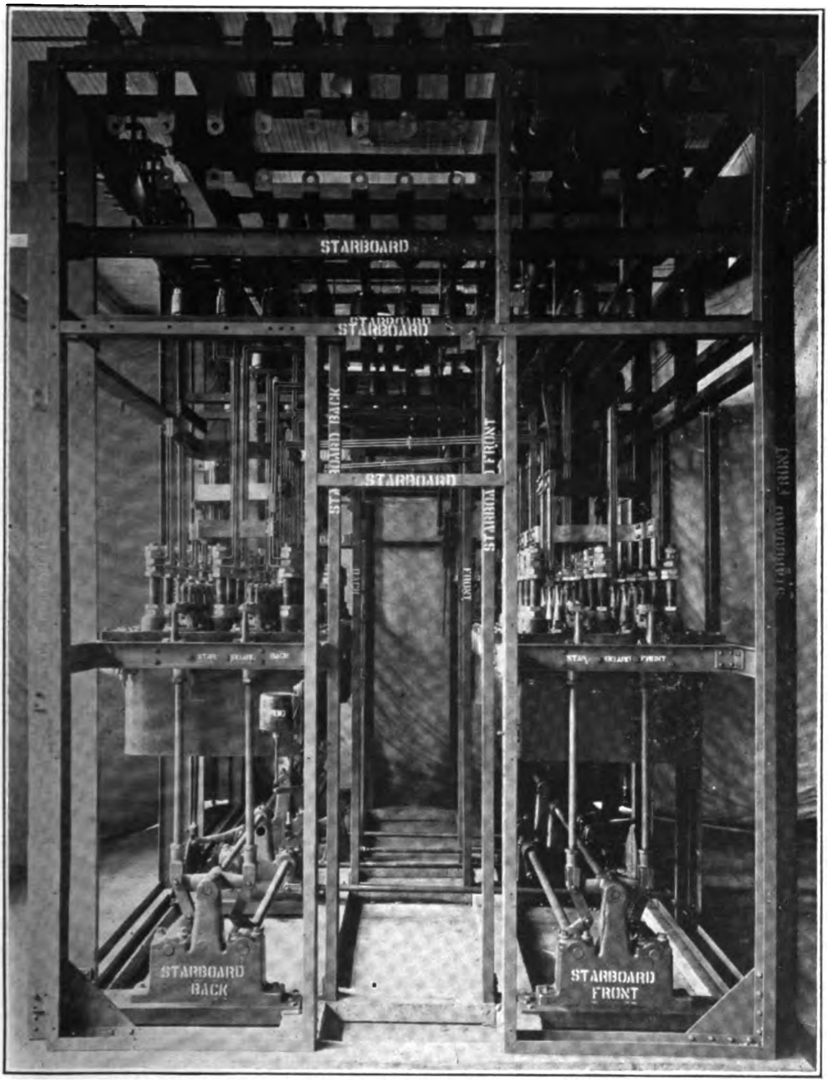


FIG. 19.—MAIN SWITCHBOARD—SIDE VIEW WITH GRILLE WORK REMOVED.

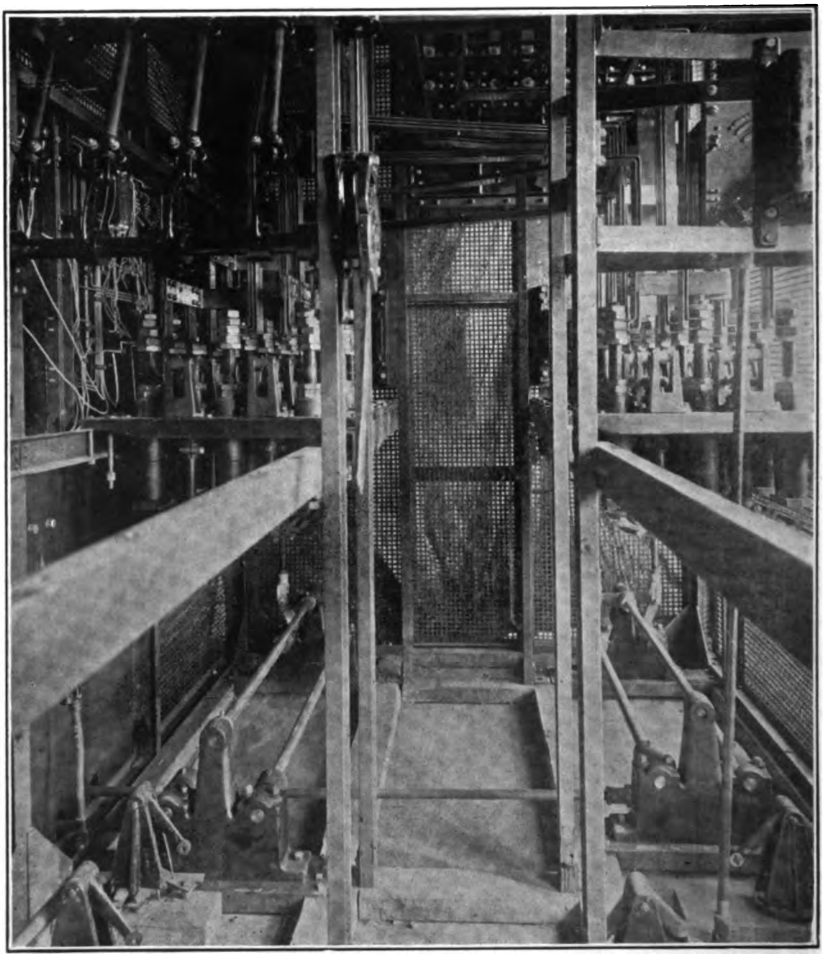


FIG. 20.—MAIN SWITCHBOARD—VIEW OF INTERIOR SHOWING INTERLOCKS AND OIL SWITCHES.

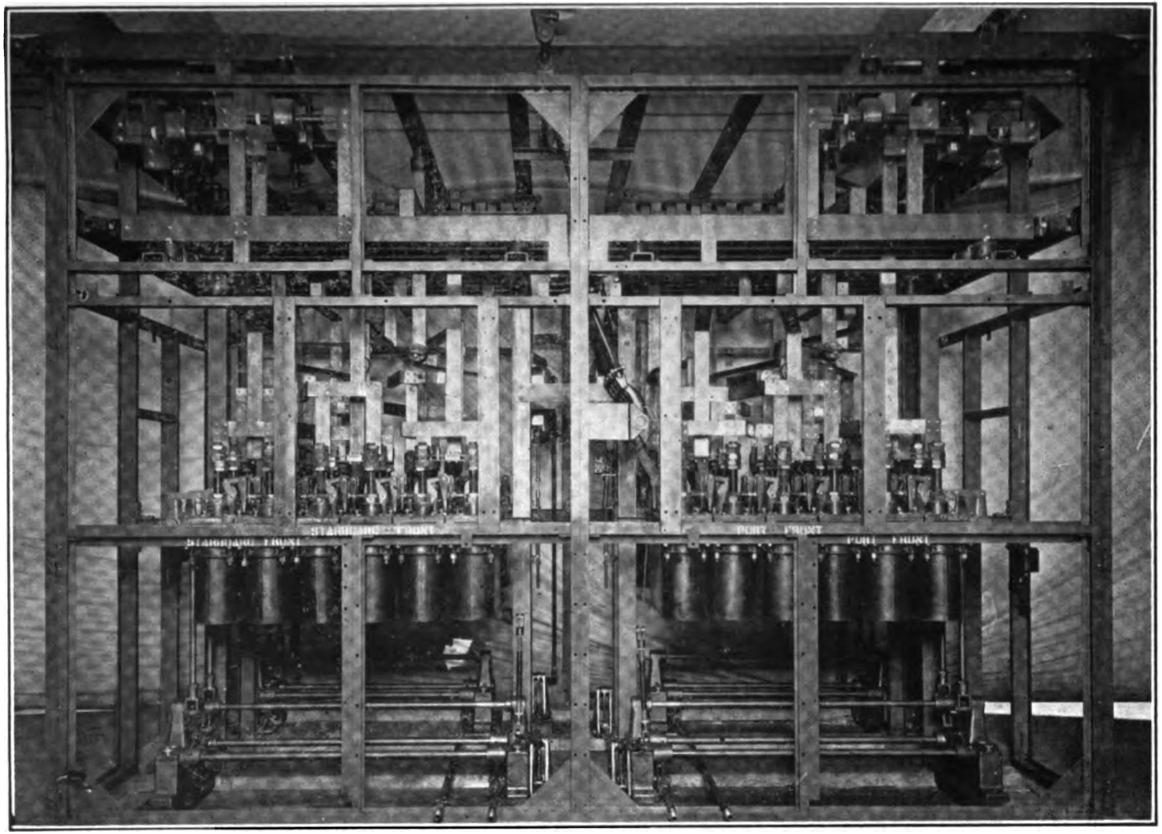


Fig. 21.—Main Switchboard—Front View with Panels Removed.

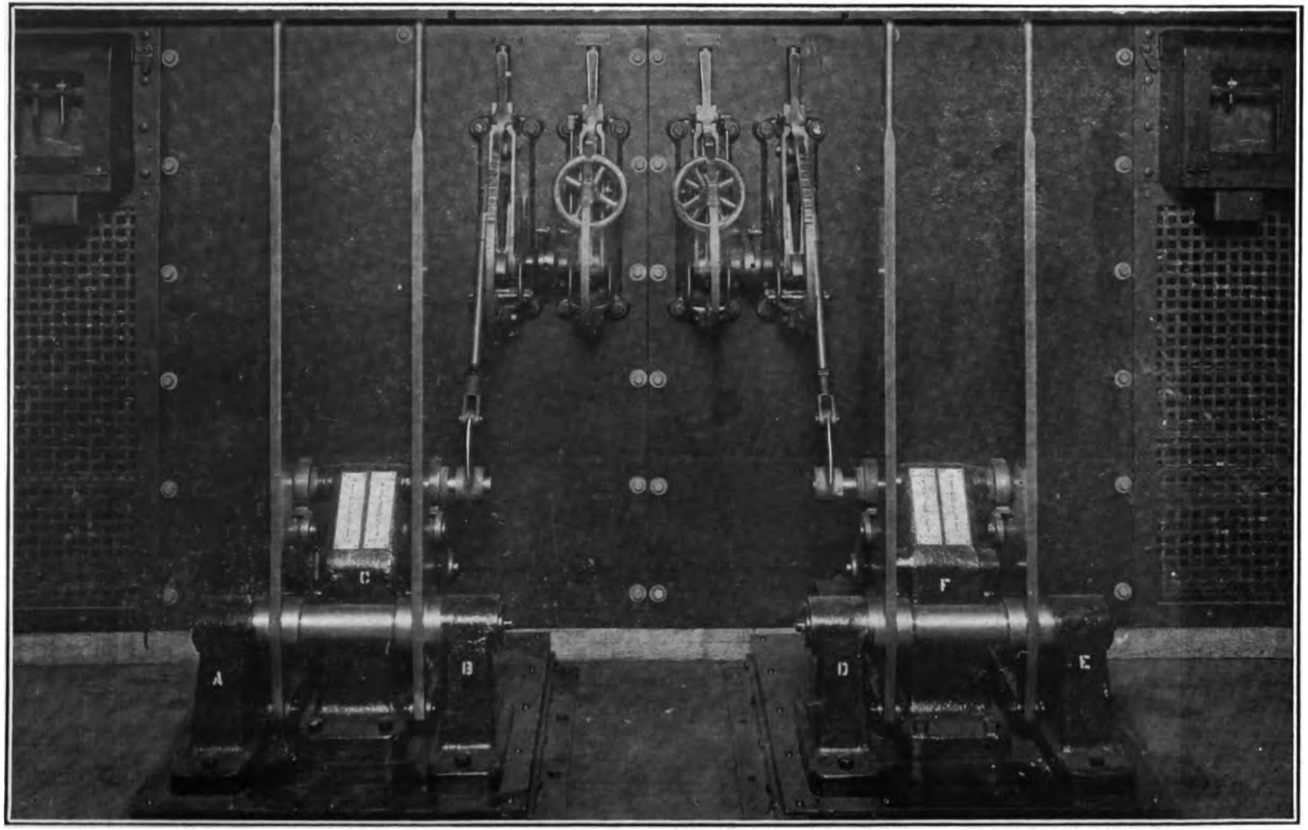


FIG. 22.—MAIN SWITCHBOARD—PARTIAL FRONT VIEW SHOWING PRINCIPAL CONTROL LEVERS.

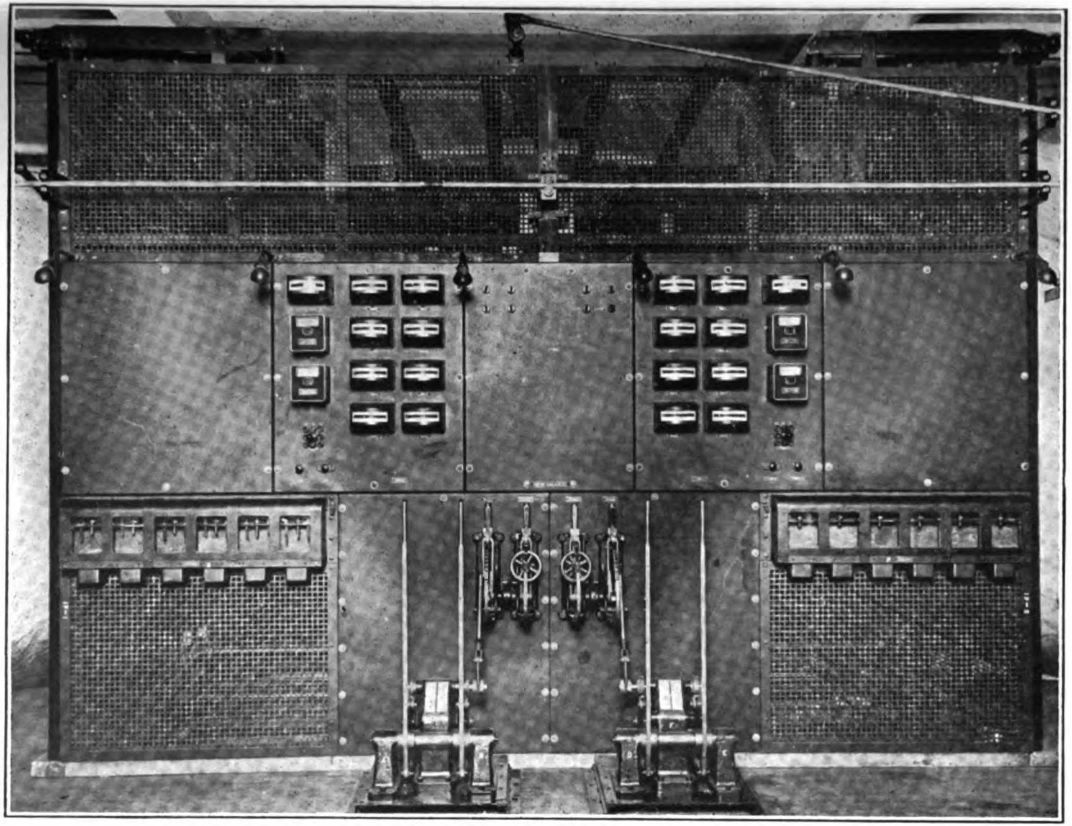


Fig. 23.—Main Switchboard—Front View Showing Levers and Principal Instruments.

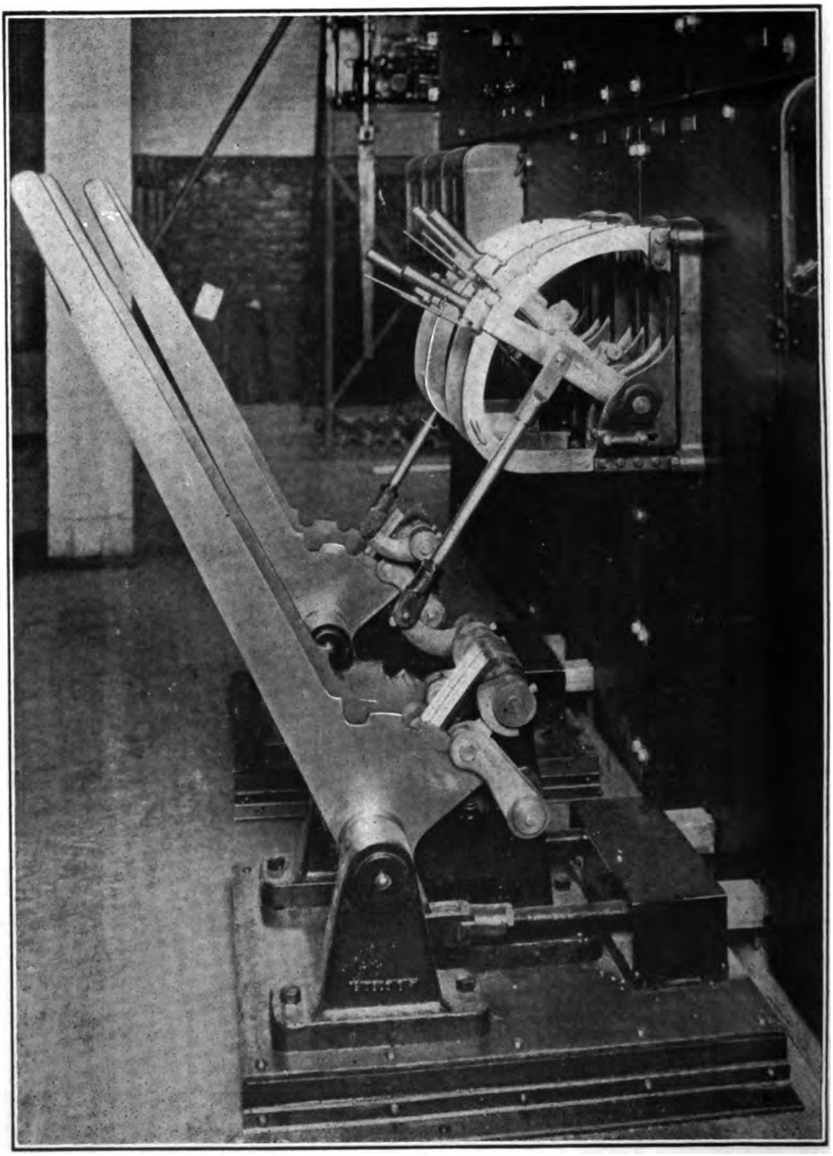


FIG. 24.—MAIN SWITCHBOARD—PARTIAL VIEW SHOWING CONTROL LEVERS.

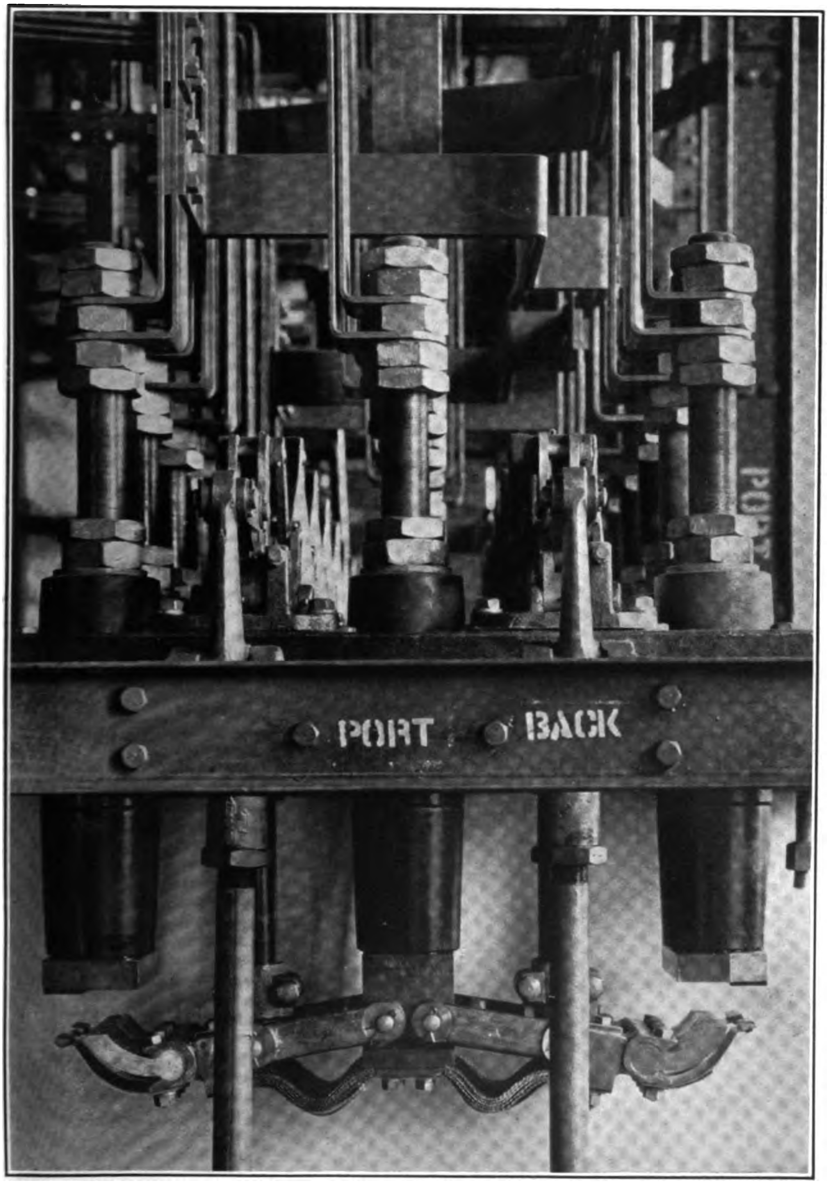


FIG. 25.—OIL SWITCH FOR REVERSING ON POLE CHANGING.

and is also provided with an auxiliary switch which short-circuits the main field through a discharge resistance, previous to opening the main field. This prevents any great rise of potential in the main field circuit due to the quick opening of the main field switch; if this resistance were not provided, opening of the main field switch would burn up the switch and also probably puncture the insulation of the main field itself. This switch is arranged for hand operation in case of failure of the solenoid-operated feature. It is also arranged so that it can be tripped out by a balanced relay located on the main switchboard and the purpose of which will be explained later on in this article.

Other switches on this switchboard aré of the knife-blade type.

Indicating lamps are placed to show which bus is alive, thus facilitating the operation of the switching and lessening chances of mistake.

The main exciter supply switches are provided with interlocks, which prevent them from being thrown in parallel on the same bus or in parallel with the ship's circuit.

The main switchboard and the main A.C. wiring and D.C. interlock wiring are shown in Figures 17 to 28, inclusive. Figures 17 to 21, inclusive, show various views of the structure of the main switchboard and of the arrangement of disconnecting switches and the interior of the board itself, together with the mechanical interlocks between switches. From these views it will be seen that the main switchboard is a cell structure, built up on angles and carrying panels for mounting instruments on its forward face. The remaining part of the switchboard is enclosed with grille work and has a door at each end which gives access to the passageway down the middle of the cell. This passageway gives access to all mechanical interlocks located inside the structure, to all oil switches, and to the bus tie switch; the latter is the only switch that is actually operated from the inside of the cell

structure. Figures 22, 23 and 24 show various views of the operating levers on the front of the switchboard. Figure 28 shows the front of the main switchboard with all instruments and levers. All the levers used in the operation of the engines are shown in these views. The switchboard is located in the after end of the center engine room, and the operator faces aft when handling the switches. By referring to Figure 27 the relation of all the switches and levers used in the various operations can be seen. There is a disconnecting switch (already described under generators) for each main generator. These are operated in the center engine room and are mounted on the outboard bulkheads on each side. These switches are of the knife-blade type; they are mounted on large brass plates which are bolted to the bulkhead and form a part of it. Single conductor cable is used to connect the generator to the main switchboard, and it is therefore necessary to replace the part of the bulkhead surrounding the cable with non-magnetic material to prevent its overheating and causing losses in the circuits. The insulators for this switch are heavy molded blocks of bakelite, which is very tough and is not susceptible to breakage by shock or vibration.

The bus tie switch is mounted inside the cell structure and is operated at that point. It is also of the knife-blade type. The bus tie switch connects the starboard and port buses so that either generator can run all four motors when desired. The bus tie switch and the two generator disconnecting switches are mechanically interlocked in such a manner that only two of them can be thrown in at any one time. This makes it impossible to connect the two generators to the same bus and thus put them in parallel. They are also mechanically interlocked so that the generator disconnecting switches can be thrown into the low-voltage position only when the bus tie switch is closed, and into the high-voltage position only when the bus tie switch is open; this interlocking makes it certain that the generators will always be operated in the most effi-

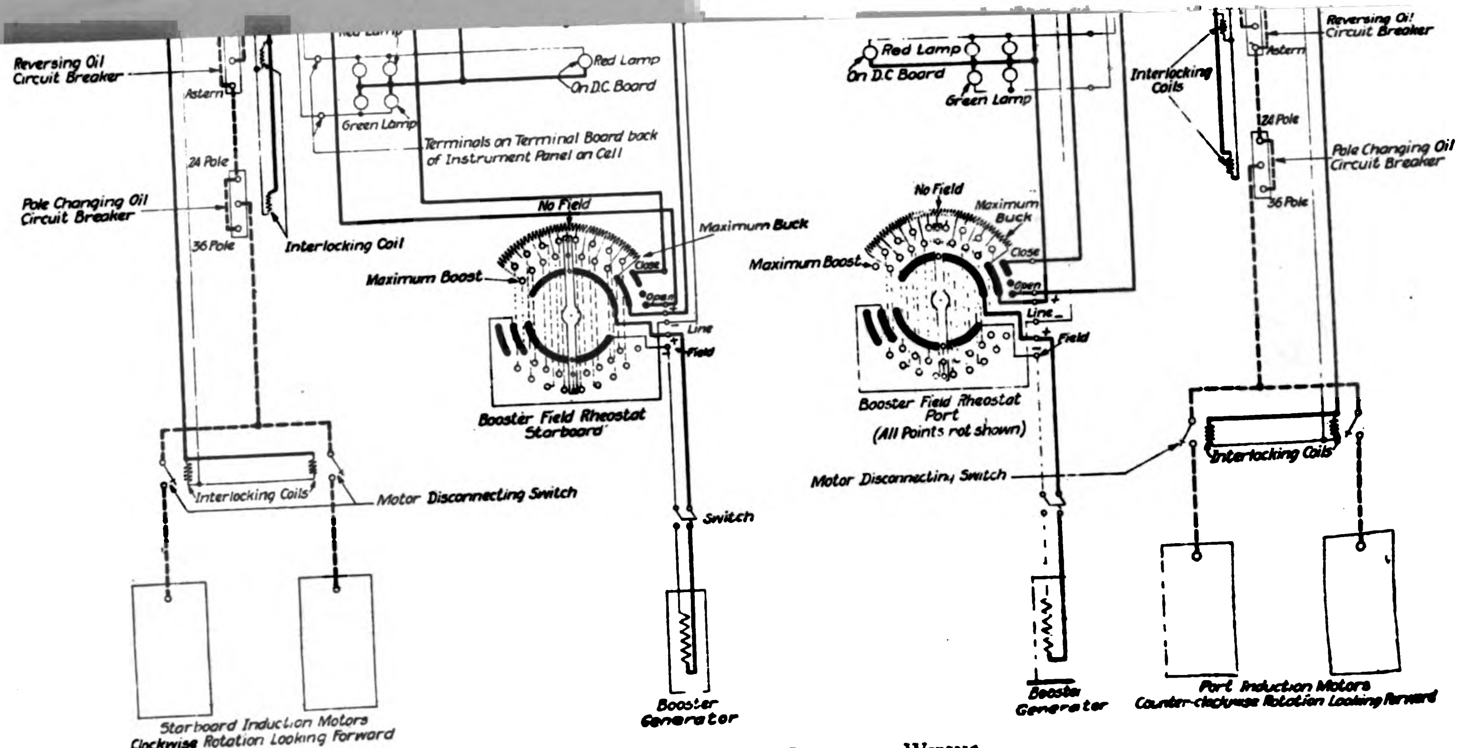
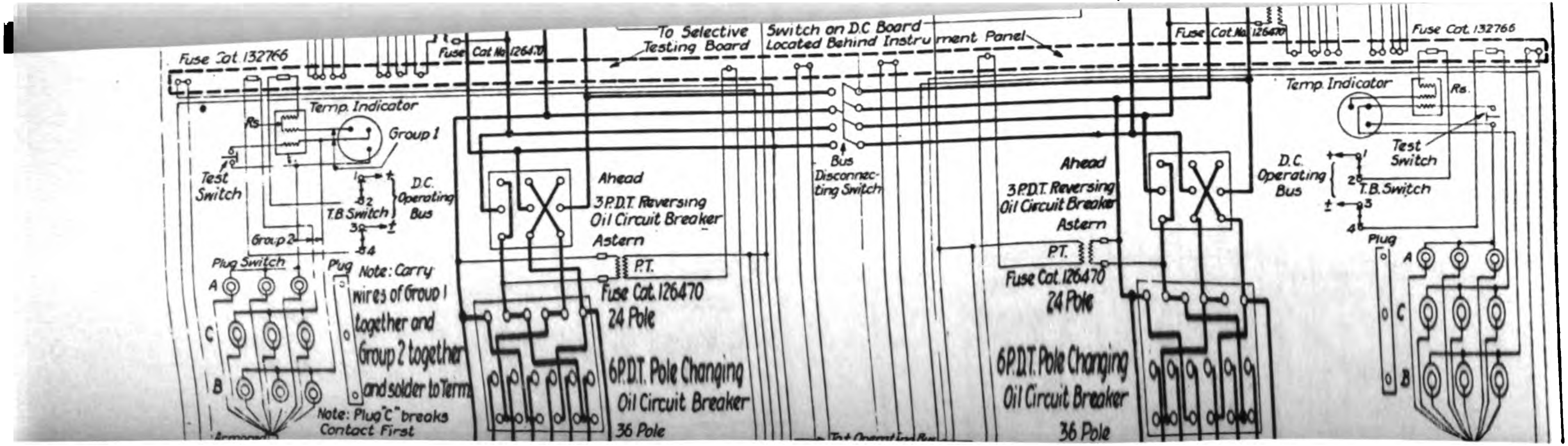


FIG. 26.—CONTROL AND INTERLOCK WIRING.


To Selective
Switch on D.C Board



Fuse Cat. 132766
Temp. Indicator
Rs
Group 1
Test Switch
D.C. Operating Bus
T.B. Switch
Group 2
Plug Switch
A
C
B
Plug
Note: Carry wires of Group 1 together and Group 2 together and solder to Term.
Note: Plug "C" breaks Contact First
Fuse Cat. No. 126470
To Selective Testing Board
Switch on D.C. Board Located Behind Instrument Panel
Ahead
3 P.D.T. Reversing Oil Circuit Breaker
Astern
P.T.
Fuse Cat. 126470
24 Pole
6 P.D.T. Pole Changing Oil Circuit Breaker
36 Pole
Bus Disconnecting Switch
Fuse Cat. No. 126470
Ahead
3 P.D.T. Reversing Oil Circuit Breaker
Astern
P.T.
Fuse Cat. 126470
24 Pole
6 P.D.T. Pole Changing Oil Circuit Breaker
36 Pole
Fuse Cat. 132766
Temp. Indicator
Rs.
Test Switch
D.C. Operating Bus
T.B. Switch
Plug
A
C
B

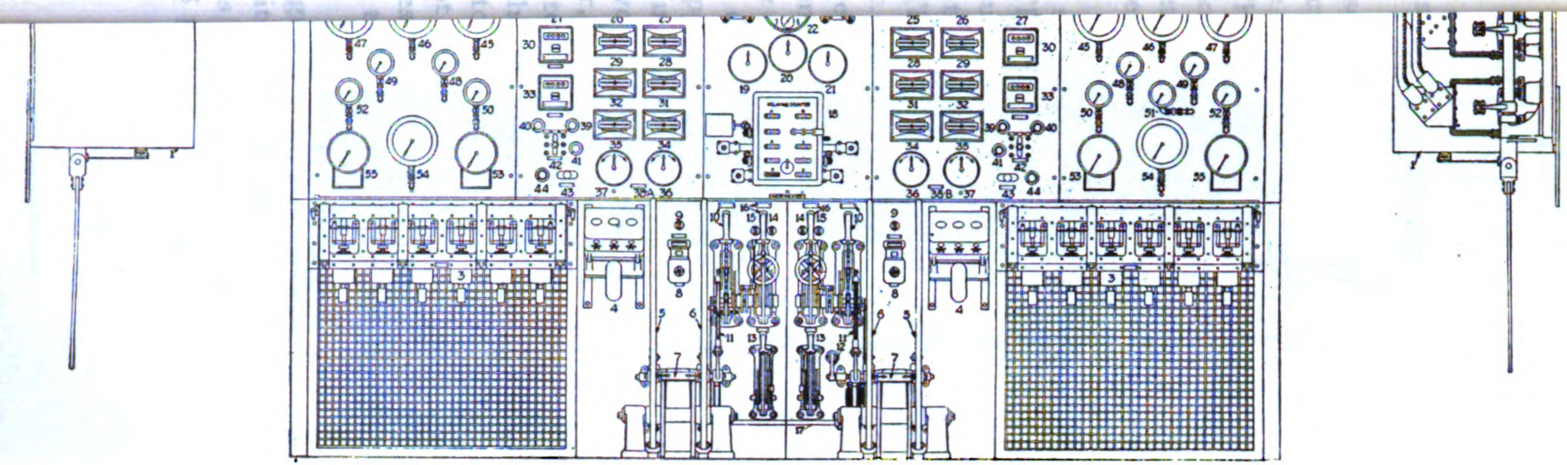


FIG. 28.—MAIN SWITCHBOARD—COMPLETE FRONT VIEW; ALSO SHOWS GENERATOR DISCONNECTING SWITCHES ON BULKHEADS.

cient way, as has already been described under the subject of generators.

The motor-disconnecting switches are mounted at the top of the cell structure on each side and at the back. These switches and the levers for operating them are shown clearly in Figure 18. These switches are also of the knife-blade type.

None of the knife-blade switches are to be operated when alive, and the means for preventing this will be explained later on in this article when the subject of electrical interlocks is discussed.

The reversing switches are of the laminated brush type, and are oil-break. They have auxiliary contacts and are intended to be large enough to break full-load currents, but in the actual operation this is never done, as will be explained under "Operation." The ahead and astern switches are mounted in the same oil tank. This switch is shown in Figure 25. It will be seen that there is a brush on each side, the two arms being pivoted at the center. The arrangement of the levers is such that one of these arms is moved up and the other down at the same time, one closing for the ahead direction and when moved in the opposite way the other closes for the astern direction. The function of this switch is merely to reverse the connection of two leads (one phase) to the motor. A third lead is, however, interrupted by this switch, as otherwise, opening this switch would still leave one phase connected to the motor, which would then run as a single-phase motor. The fourth lead to the motor passes direct from the bus to the motor and is not interrupted by the reversing switch. The reversing switches for the two motors on one side of the ship are operated by the same lever on the front of the switchboard.

The pole-changing switches are in every respect similar to the reversing switches. Contact of the laminated brushes on one side connects the motor for 24 poles and contact on the other side for 36 poles. The switches for the two motors on

one side of the ship are operated by one lever on the front of the switchboard.

The pole-changing and reversing switches are mechanically interlocked with the field lever so that they cannot be moved unless the field lever is in the open position. They are also interlocked with each other so that the reversing lever cannot be thrown in the astern position except when the pole-changer is in the 36-pole position. This latter interlock is necessary for reasons described under the subject of the "main motors."

The main field switches have already been described under the subject of the exciter switchboard; they are operated by the outboard small lever shown in the center of the main switchboard.

The speed levers are the two small inboard levers at the center of the main switchboard. They are connected by bell cranks and rods to the governor-control gear, as shown in Figure 2 and previously described under the subject of the "main turbine." These levers move on notched quadrants; they are moved from notch to notch when comparatively large changes of speed are quickly desired, but small changes of speed are made by vernier wheels. This type of lever has not proved very satisfactory in service, however, and a new type is being installed; this new type is similar to the one installed on the *Jupiter,* and consists of a worm and worm-wheel transmission which enables the operator to go through the full range of speed without lifting the lever from the quadrant; in addition, it will be possible to lift the worm out of engagement and move it in either direction when large changes in speed are quickly desired. The speed lever and field lever are mechanically interlocked so that the speed lever is automatically carried to the slow-speed position when the field switch is open. Also, the speed cannot be increased very greatly without carrying the field lever with it and increasing the excitation on the main field of the generator. This prevents dropping motors out of step by large increases of speed of the generator without corresponding increases in its excitation.

The steam limit levers are mounted directly underneath the field and speed levers and are similar to them in type, consisting of short arms moving on notched quadrants. The motion of the steam-limit levers is transmitted by rods and bell cranks to a stop on the operating gear of the main governor. This lever can be set to allow any desired number of valves to open on the main turbine and when once set the amount of power that can be developed by the turbine is limited by the amount of steam that the given number of valves will pass. If the load becomes such that the required amount of steam to maintain the speed is greater than that allowed by the steam limit, the turbine will simply slow down as would any other type of throttled engine. The relative position of the steam-limit stop and governor-operating mechanism is always shown at the main switchboard, as will be explained under the subject of electric interlocks. The necessity for the steam limit arises from the conditions that obtain when the ship makes a turn. In this case, the power required to maintain the revolutions the same as they were before the turn is enormously increased; since the turbine is at all times under the control of the governor, it will continue to maintain the same speed while the ship is turning that it did before the turn started, and this will result in overloading the main generator beyond its capacity; this will result in a drop in its voltage and the main motors will then drop out of step due to insufficient torque. But when the steam limit is set, the turbine will slow down before the excessive overload is reached on the generator, and the latter will then have sufficient voltage to furnish the necessary torque for holding the motors in step.

The D.C. interlocks, booster-field control, and main control wiring are shown in Figure 26. The booster is a motor generator set, the motor end of which is a constant-speed motor and the generator end of which has a separately excited reversible field. The field current for excitation of the main

generator passes in series through the booster generator and regulation of the strength of the main field current is obtained by varying the voltage of the booster generator. This either bucks or boosts the voltage of the 300-kw. generator, according to the throw of the field lever. The arrangement of the field windings and the booster-field rheostat is shown in Figure 26. The range of the booster generator is about 60 volts in either the buck or boost direction. Generally the boosting is only used when it is desired to put over-excitation on the main field of the generator during reversal.

The various indicating lamps and electric interlocks used in the operation of the machinery are shown in Figure 26. These are supplied by a double-throw transfer switch and a double-throw ordinary knife-blade switch. The transfer switch supplies current either from the exciter bus or from the after distribution room. In transferring from one to the other the circuit is not interrupted, but resistances are inserted directly in the two circuits so as to make it safe to transfer from one to the other without paralleling the generators. The supply bus from the exciter is fed through a double-throw switch which can be thrown on either exciter.

There is a balanced-relay in each main generator circuit which is shown at the top of Figure 26. This relay is normally in the open position and is held rigidly so by the solenoids in each circuit. In case of short-circuit on either of the phases of the generator itself, the motors, or the cable, the pull on the solenoids of this relay will become unbalanced, and the contacts in the D.C. circuit will be closed, tripping out the main field switch and taking all power off the generator in question.

Under-load relays are also provided for each generator circuit. The solenoids of these relays are adjustable and can be made to open at any desired value. Until this relay opens, the reversing and pole-changing levers are locked so they cannot be moved. A blue lamp indicates when this circuit is closed.

The main field switch, when closed, electrically locks the main generator disconnecting switches, motor disconnecting switches, and the doors to the cell structure of the main switchboard. This insures that none of these switches will be moved while the circuits are alive. There are red lamps for indicating when the switch is closed and green lamps for indicating when it is open. When only one generator, and consequently one field switch, is in use, the bus-tie switch will always be closed. The bus-tie switch is therefore provided with an auxiliary switch which connects the two interlocking systems on the two sides of the ship so that either field switch will then operate all interlocks.

White and red lamps are provided on each side of the board to show the relative position of the steam-limit stop and the main governor-control gear. The white light indicates an opening of about ⅛ inch (this is adjustable) between the stop and governor gear. When both lamps are lighted this indicates contact between the stop and governor gear, and when the white light goes out, leaving only the red lamp on, it indicates that the governor gear is hard up against the stop. This enables the operator to keep his steam limit properly adjusted for each speed of the ship.

The *New Mexico* is operated from low speeds up to 15 knots using one generator, and the motors connected for the 36-pole connection; from 15 knots to 17 knots one generator is used with the motors on the 24-pole connection; from 17 knots to full power, both generators are used and motors will be arranged for the 24-pole connection. There are thus three conditions of normal steaming and the operation of the ship for both ahead and astern motion will be described for each of these conditions. The operation is as follows:

(1) *Getting under way with one generator and motors on 36-pole connection.*—Assume that the maximum speed will not be over 15 knots. This means that one generator will be used and that the motors will be operated on the 36-pole con-

nection. Before reporting ready to get under way, the generator that it is desired to use would be tested out and then set to run at low speed; the bus-tie switch would be closed, and the generator-disconnecting switch would be closed in the low-voltage position. The field switch of the generator and the pole-changing and reversing levers would be open. On receiving a signal "Ahead,"

(a) The pole-changing switch would be thrown in the "36-pole" condition.

(b) The reversing lever would be thrown in the "Ahead" position.

(c) The field switch would be closed and brought up to the desired field strength.

(d) The turbine would then be brought up to the desired speed.

(2) *Reversing.*—With the ship going ahead under conditions described above, under (1), on receiving a signal "Astern,"

(a) The field switch would be opened and the speed lever brought to low speed at the same time.

(b) As soon as the under-current relay operates, the reversing levers would be thrown to the "Astern" position.

(c) The field switch would then be closed and over-excitation applied to the generator until the motors have reversed and come up to synchronous speed, when the field would be reduced to its normal value.

(d) The speed of the turbine would then be brought up to whatever is desired.

(3) *Getting under way with one generator and motors on 24-pole connection.*—In case it is desired to make the maximum speed greater than 15 knots, but less than 17½ knots, one generator will be used and the motors will be connected on the 24-pole condition. In this case, getting under way is exactly as described under (1), except that the pole-changing switches would be thrown into the 24-pole position.

(4) *Reversing.*—Assuming the ship to have gotten under way with conditions as described above, under (3), on receiving a signal " Astern,"

(a) Open the field switch and throw the speed lever to low speed at the same time.

(b) Throw pole-changers to 36-pole position.

(c) Throw reversing switches to " Astern" position.

(d) Apply over-excitation to the field of the main generator until motors have come into synchronism and then reduce field to normal value.

(e) Bring turbine up to desired speed.

(5) *Getting under way with two generators.*—In this case the bus-tie switch would be open and the two generator-disconnecting switches would be closed in the high-voltage position. Field switches would be open; generators would be running at slow speed and pole-changing and reversing switchs would be open. If signal "Ahead" is received, the operations carried out would be exactly as described under (3), except that it would be performed on two generators instead of one.

(6) *Reversing.*—After having gotten under way with conditions as above described under (5), if a signal " Astern" were received, the operations would be exactly as described under (4), except that they would be carried out for two generators instead of one.

SHAFTING AND BEARINGS.

There are four lines of shafting, each driven by one of the main propelling motors. Each line of shafting is composed of the following number of separate sections, all rigidly coupled together:

	Inboard shafts.	Outboard shafts.
Thrust shaft	1	1
Line shaft	2	1
Stern-tube shaft	1	1
Propeller shaft	1	1

Thrust Shafts.—The thrust shafts are fitted to thrust bearings of the adjustable horseshoe type, with steady bearings at each end for supporting the shafts. The shafts are hollow, with thrust collars and coupling discs forged integral with the shafts.

Line Shafts.—These shafts are hollow forged, with disc couplings at either end, and each is carried by two steady bearings.

Stern-Tube Shafts.—The stern-tube shafts are covered with a composition casing within the stern tubes and at the stern tube bearings. They are hollow forged, secured to the line shafts at inboard end by special disc couplings, and to the propeller shafts at outboard end by sleeve couplings. Two lignum vitæ lined bearings are provided for each stern tube shaft.

Propeller Shafts.—Each propeller shaft is carried by one lignum vitæ lined strut bearing. The shafts are hollow forged, taper turned at after end to suit the propeller hub, and composition bushed at the bearings.

Inboard Coupling.—Each inboard coupling consists of a sleeve secured to the stern-tube shaft by four keys, the end of the shaft being slightly upset for reception of the sleeve. Back of the sleeve is a collar made in halves and secured to the sleeve and to the coupling disc on the line shaft by fitted bolts. The shaft is prevented from pulling out of the sleeve when backing by the half collars, which shoulder against the upset on the end of the shaft.

Outboard Coupling.—Each outboard coupling is of the solid-sleeve type, taper bored for reception of shafts, and secured to each shaft by two feather keys and one cross key.

SHAFT DATA.

Thrust shafts, length, feet and inches	16–06
diameter, inches	15¼
at bearings, inches	15½
axial hole, inches	7⅝
collars, number	12

Thrust shafts, collars, thickness, inches	2
space between, inches	4
outside diameter, inches	23
inside diameter, inches	15½
Line shafts, forward, inboard, length, feet and inches	29–03
after, inboard, length, feet and inches	29–03
outboard, length, feet and inches	18–06
diameter, inches	15¼
at bearings	15½
axial hole, inches	7⅝
Stern-tube shafts, length, feet and inches	42–02⅝
diameter, inches	16⅞
axial hole, inches	8⅜
Propeller shafts, length, feet and inches	34–11¾
diameter, inches	16⅞
axial hole, inches	8⅜
Coupling discs, diameter, inches	30
thickness, inches	3½
Inboard coupling, diameter of sleeve outside, inches	30
inside, inches	16⅞
length of sleeve, inches	13
thickness of collar, inches	3½
Coupling bolts, number each coupling	8
diameter (taper) at face of coupling, inches	3¼
Outboard couplings, length of sleeve, inches	63¼
diameter of sleeve, inches	22½

BEARING DATA.

Thrust bearings (white-metal lined bearings and shoes):	
Steady bearings, number each	2
diameter, inches	15½
length, inches	19
Thrust shoes, number each	12
effective surface, square inches, ahead	1,782
astern	1,635.5
Line-shaft bearings (white-metal lined):	
diameter, inches	15½
length, inches	26
Stern-tube bearings (lignum vitæ lined):	
forward bearing, diameter, inches	19⅛
length, inches	48
after bearings, diameter, inches	19⅛
length, inches	60
Strut bearings (lignum vitæ lined):	
diameter, inches	19⅛
length, inches	72

TURNING GEAR.

Each line of main shafting is provided with a motor-driven, reversible turning gear. The motors are 5-horsepower each. Each gear is so arranged that it can be operated by hand, a long ratchet wrench being provided for that purpose.

PROPELLERS.

There are four three-bladed propellers, with the blades cast solid with the hubs. The starboard propellers are right-hand and the port propellers are left-hand; all propellers turn outboard when driving the vessel ahead. They are of manganese bronze and the hubs are taper bored to fit the shafts, to which they are secured by a key and locked nut on the end of each shaft. The blades are machined true to pitch, and polished all over, including the hubs.

PROPELLER DATA.

Diameter of propeller, feet and inches	13–5
hub, feet and inches	2–9
Pitch, feet and inches	15–2
Ratio of diameter to pitch	0.8846
Area, projected, square feet	50.43
helicoidal, square feet	61.22
disc, square feet	141.38
Ratio, projected to disc area	0.3567
helicoidal to disc area	0.4333
Heighth of lower tip of blade above keel, inches, inboard	11⅛
outboard	20⅛
Immersion of upper tip of blade, feet and inches, inboard	15–07⅞
outboard	14–10⅞

MAIN CONDENSING APPARATUS.

Main Condensers.—There are two main condensers, one for each main turbine. They are installed in the same compartments with the main turbo-generators and supported by substantial foundations above the turbines. The tubes are straight, rolled into the tube sheets at one end and gland packed at the other end. The condensers are of cylindrical

cross-section and conforms to Navy Standard practice. The principal dimensions follow:

Inside diameter, feet and inches	10–3
Thickness of shell (steel), inch	0½
Length between tube sheets, feet and inches	10–6
Thickness of tube sheets, inches	1⅜
Tubes number	8,901
diameter, outside, inch	0⅝
thickness, inch	0.065
Cooling surface, square feet	15,300
Size of main exhaust nozzles, inches	56½×75
Diameter of air-pump suction, inches	10
condensate pump suction, inches	6
auxiliary exhaust nozzle, inches	10
circulating-water inlet and outlet, inches	26

Dry Air Pumps.—A vertical, motor-driven, direct-connected, double-acting, single, dry-vacuum pump is provided for each main condenser. They are located in the center engine room and are arranged to draw air and non-condensible vapors from the condensers and discharge same to the atmosphere above the main deck. The pump suctions are cross-connected so that either pump may be used on both condensers in emergencies. The pumps and motors are of the following principal characteristics:

Make of pump	Wheeler Condenser & Engineering Co.
Diameter air cylinder, inches	30
Stroke, inches	18
R.p.m.	100
Diameter suction, inches	10
discharge, inches	6
Horsepower of motor	40
Make of motor	G. E. Co.

Condensate or Hotwell Pumps.—The condensate is drawn off the bottom of each main condenser by a horizontal motor-driven, direct-connected, double-inlet centrifugal pump discharging into the filter chambers of the feed and filter tanks. The pumps are located in the center engine room and have a cross-connection in their suctions so that either may be used on both main condensers, as is the case with the air pumps.

The pumps and motors are of the following principal characteristics:

Make of pump	Alberger Pump & Condenser Co.
Capacity, gallons per minute	500
Diameter impeller, inches	9½
suction, inches	8
discharge, inches	6
Horsepower of motor	20
R.p.m.	1,800
Make of motor	G. E. Co.

Main Circulating Pumps.—Circulating water for each main condenser is supplied by a horizontal motor-driven, direct-connected, double-inlet, centrifugal pump located in the center engine room. They are also arranged for pumping on the engine-room bilges in emergencies. The pumps and motors are of the following principal characteristics:

Make of pump	Alberger Pump & Condenser Co.
Capacity, gallons per minute	21,200
Diameter, impeller, inches	26
sea suction, inches	26
bilge suction, inches	21
discharge (at pump), inches	24
Horsepower of motor	250
R.p.m.	500
Make of motor	G. E. Co.

FORCED—LUBRICATION SYSTEM.

The main turbo-generator sets, the main propelling motors and the thrust bearings are provided with forced lubrication.

The pumping equipment, located in the center engine room, consists of four motor-driven, geared, oil pumps of the Kinney type, four Schutte-Koerting, type Z, oil coolers (two No. 5, of 127 square feet of cooling surface each, and two No. 7, of 212 square feet of cooling surface each), and two 7-inch by 8-inch by 12-inch vertical, piston, double-acting, single, Blake pumps for supplying cooling water to the oil coolers. Two oil drain tanks of 1,000 gallons' capacity each and four discharge strainers are also provided.

The lubricating-oil pumps draw oil from the drain tanks and discharge same through pressure reducing bafflers to the oil coolers, thence, via the lubrication main and its branches, to the bearings. The oil coolers can be by-passed if so desired. The oil from the bearings is led back to the drain tanks through drain lines, thus completing the cycle, which is indefinitely repeated. Cut-out valves are provided as necessary for isolating units not in operation.

A thermometer is fitted at each oil-cooler inlet and outlet, and also at each bearing.

Oil for the main turbine-governor gear is also supplied by the lubricating-oil pumps at a pressure of about 80 pounds per square inch. In order to keep this excessive pressure off the oil coolers and lubrication lines, bafflers are placed in the pump discharges beyond the branches to the governor gear, which reduce the pressure on the lubricating system to about 40 pounds per square inch.

Reserve lubricating oil is carried in two storage tanks of 3,000 gallons combined capacity. These tanks are provided with deck-filling connections and are arranged for readily replenishing the forced-lubrication system.

Two 1,000-gallon lubricating-oil treating tanks, fitted with steam heating coils, are also provided. The forced-lubrication pumps are arranged to pump the oil from the system into these tanks when it requires cleansing.

WATER SERVICE SYSTEM.

The water service for the main turbines, thrust and spring bearings and the stern tubes is supplied from connections taken off the main circulating pumps, fire and bilge pumps and oil-cooler circulating pumps discharges. There is also an emergency connection from the fire main.

Branches, controlled by separate valves or cocks, are led to the various bearings and auxiliaries requiring cooling. Hose valves are also provided at convenient locations.

The drains from the main turbine bearings discharge into funnels so that they can be readily observed. The funnels are piped to a small drain tank in the center engine room, which has connections to the fire and bilge-pump suctions for pumping overboard. All other drains are closed circuits, uniting in a common main, and discharge overboard through the auxiliary overboard discharge sea chest on the center engine room.

FIRE AND BILGE PUMPS.

There are seven fire and bilge pumps, four located in the center engine room and one in each fireroom.

They have suctions from the sea and the drainage systems, and discharge to the fire and flushing mains and overboard. The engine-room pumps also have a suction from the water-service drain tank and a discharge to the oil coolers and water service.

The pumps are of the following characteristics:

Type—Worthington, vertical, piston, double-acting, single.
Size—12 inches by 10 inches by 18 inches.
Diameter of suction and discharge, 5 inches.

BOILERS.

There are nine Babcock & Wilcox water-tube boilers, arranged three abreast in three separate watertight compartments.

The boilers have a total heating surface of 55,458 square feet, exclusive of superheating surface, and are designed to operate the entire machinery plant at full power, with an average air pressure in the ash pits of not more than 6 inches of water. They are equipped for burning fuel oil.

Uptakes of the usual design are provided, all uniting into a common smoke pipe at the main deck. The smoke pipe is 14 feet 3 inches inside diameter, and 92 feet 6½ inches high above the bottom of the furnaces.

BOILER DATA.

Number	9
Pressure, working, pounds per square inch	280
test, pounds per square inch........water, 420; steam	330
Height to top, external, feet and inches	15 1 11/16
Length on floor, feet and inches	9–0¼
Width on floor, feet and inches	17–3⅝
Drum, diameter, inside, inches	42
length, feet and inches	18–0¼
thickness, inch	11/16
Number of tube headers, each boiler	56
Distance between headers, feet and inches	9–00
Number of tubes, each boiler:	
2-inch generating tubes, 134 mils. thick	1,210
4-inch generating tubes, 165 mils. thick	28
2-inch superheater tubes, wing boilers, 134 mils. thick	46
2-inch superheater tubes, wing boilers, 165 mils. thick	46
2-inch superheater tubes, center boilers, 134 mils. thick	54
2-inch superheater tubes, center boilers, 165 mils. thick	54
Number of furnaces, each boiler	1
Furnace volume, each boiler, cubic feet	567
Heating surface, each boiler, square feet:	
Generating tubes	6,162
Superheater tubes, wing boilers	470
center boilers	552
Area through smoke pipe, square feet	149.5
$\frac{\text{Area through smoke pipe}}{\text{Furnace volume}}$	0.0293
Kind of forced draft	Closed firerooms.
Atomizers, number each boiler	7
type	Peabody mechanical atomizers.
Diameter of main steam stop valve, inches	7
main feed stop and check valves, inches	3½
auxiliary feed stop and check valve, inches	3
surface-blow valves, inches	01½
bottom-blow valves (two), inches	01½
safety valve (two), duplex, inches	04

FUEL–OIL SYSTEM.

The fuel-oil tanks are filled through eight 6-inch connections, two on each side of the ship forward and two on each side of the ship aft.

The two starboard forward filling connections are combined into an 8-inch pipe, which is led to a relay tank located on the third deck, starboard side. The port forward filling pipes are similarly connected to the same relay tank. There is another relay tank aft, located on the third deck, port side; the after filling connections are similarly led to this relay tank.

From each relay tank two 8-inch pipes are led to the distributing manifolds in the center engine room and forward firerooms.

In addition to the main filling connection described above, there are four emergency filling connections, two starboard and two port, each connecting to one of the four distributing manifolds in firerooms Nos. 1 and 3.

From the bottom of each fuel oil tank 4½-inch pipes, as required, are led to the various distributing manifolds, which are inter-connected so that any pump may be used on any tank. Each tank is fitted with a high suction and a low suction.

Two of the oil tanks, just forward of fireroom No. 1, are used as service tanks; similarly two others, just aft of fireroom No. 3. The fuel oil for the burners is normally taken from these tanks, which are replenished from the storage tanks as described elsewhere, but the system is so arranged that oil may be used from any storage tank direct.

Two 7-inch by 8-inch by 12-inch Blake, vertical, piston, double-acting, single, light-service booster pumps are installed in each fireroom Nos. 1 and 3. The speed of each pump is controlled by a Foster pressure governor. Each pump has a 4½-inch suction connection from the adjacent distributing manifold, and a 4-inch discharge to the distributing manifold, adjacent emergency deck-filling connection, fuel-oil main and overboard; the latter connection is portable. The booster pumps are also arranged to pump from the storage tanks and discharge to the service tanks while the service pumps are pumping from the service tanks. In addition to the connections noted above, the booster pumps also have a discharge to

the deck for fueling the launches and motor boats; also a branch to the galleys and foundry.

There are two heavy-pressure, horizontal, turbo-driven, rotary service pumps of the Kinney type in each fireroom. Mason pressure governors are provided to control the speed of the units; the turbines are also provided with centrifugal, emergency-trip governors to prevent overspeeding. They have 3-inch suctions from the fuel-oil main and service tanks, and each discharge, through an oil meter and an oil heater, to the atomizer supply pipes. Small by-passes are provided between the discharge and suction sides of the pumps for use when only a small amount of oil is being burned.

The oil meters are of the Bowser type, and are arranged so that they can be by-passed.

The oil heaters are of the Griscom-Russell type, each of sufficient capacity to heat 14,000 pounds of oil per hour from 70 degrees to 240 degrees F., with steam at 250 pounds' pressure. The oil heaters are arranged so that they can be by-passed.

The atomizers, seven per boiler, are carried in one row on the boiler fronts, with stop cocks at each burner for individual control. They are of the Peabody mechanically atomized type. The atomizer-supply pipe to each boiler is fitted with a cutout valve.

Steam coils are fitted near the ends of all suction pipes in all tanks; steam connections for boiling-out the tanks are also provided.

For raising steam with no air pressure available, a 1½-inch by 3½-inch duplex hand pump is fitted in each fireroom, capable of supplying oil at 200 pounds' pressure to two atomizers.

PNEUMERCATOR SYSTEM.

For ascertaining at any time the amount of fuel oil in the various double-bottom tanks, the Parks pneumercator system is installed.

The installation consists of a small semi-spherical balance chamber placed at a predetermined location near the bottom of each tank. The interior of the balance chamber is in communication with the tank, through a hole in the side of the chamber, and a small pipe connects the top of the chamber with its recording instrument in the engine rooms and firerooms at the fuel-oil manifolds. The recording device consists of a glass mercurial tube provided with scale, calibrated to suit the tank to which it is connected.

The instrument operates on the following principle: The pressure due to the head of oil in the tank compresses the air in the chamber and connecting pipe line, which in turn causes the mercury to rise or fall in the tube in direct ratio to the pressure exerted. The tank contents is read off on the scale, as in the case of an ordinary thermometer.

In order to guard against overflowing the tanks when taking aboard fuel oil, an annunciator is installed in connection with the recording device for indicating and signalling when any tank is 95 per cent. full.

FORCED DRAFT BLOWERS.

Forced-Draft Blowers.—Each fireroom has three forced-draft blowers. They are located in specially constructed blower rooms just below the third deck and above the working flat in front of the boilers.

The fans are 17½ inches outside diameter, Keith, horizontal, double-inlet type, each capable of delivering 23,000 cubic feet of air per minute against a static pressure of 8 inches of water, when running at 1,800 revolutions per minute.

Each fan is direct-connected, by a flexible coupling, to a horizontal Terry steam turbine.

Air is supplied the blowers from a large air duct on the second deck.

The blowers discharge the air into the firerooms through short trunks, extending through openings in the blower-room floor.

Deflectors are fitted at the trunk outlets, so arranged that the air will be properly diffused throughout the firerooms.

An automatic hinged shutter, closing by back pressure, is fitted in each blower discharge to prevent the loss of air pressure in the fireroom in case a blower stops.

MAIN STEAM PIPING.

The main steam piping is arranged in two symmetrical systems, one on each side of the vessel. The two lines are cross-connected in each fireroom. The branches from the boilers are 7 inches in diameter each, and the lines proper are 8½ inches in the forward fireroom, increasing to 11, 12 and 14 inches at each successive boiler connection, the latter size continuing to the main engine's throttle valves. The piping is seamless-drawn steel.

AUXILIARY STEAM PIPING.

The auxiliary steam pipe is seamless-drawn steel.

From the main steam pipe in the after fireroom there are two 6½-inch connections leading aft through the after dynamo room and forming a loop in the center engine room. The engine room auxiliaries take their steam from this loop.

In the firerooms the auxiliary steam piping consists of branches from the main steam piping with sub-branches to the various auxiliaries.

The ship's turbo-generator sets take their steam off the main steam cross-connection in the firerooms Nos. 1 and 3.

Stop valves are fitted in all branches and sub-branches,

AUXILIARY EXHAUST PIPING.

A single copper auxiliary exhaust pipe, ranging in size from 10 inches to 14 inches, is led throughout the machinery spaces with branches leading elsewhere as required for the various auxiliaries. Connections are provided to direct the exhaust steam into either main condenser, either dynamo condenser,

either feed-water heater, evaporators, or into the atmosphere through the escape pipe at will.

Stop valves are fitted in all branches at the main.

ESCAPE PIPES AND ATMOSPHERIC EXHAUST.

There are two 15½-inch escape pipes, one forward and the other aft of the smoke pipe. Branches from these pipes are led through the third deck to each fireroom, where they are further subdivided and connected to the individual safety valves through 6-inch branches.

There is a 10-inch branch in firerooms Nos. 1 and 3 from the auxiliary exhaust to the escape pipe for atmospheric exhause. A stop valve is fitted at each connection to the escape pipe.

There is also a 10-inch atmospheric exhaust connection, fitted with automatic valve, from each dynamo turbine to the escape pipes.

All the above piping is copper.

MAIN AND AUXILIARY FEED SYSTEMS.

Feed and Filter Tanks.—Two feed and filter tanks of 5,200 gallons' capacity each, are located in each center engine room. A filter chamber of about 900 gallons capacity is in the top of each tank. Each filter chamber has an inner bottom of loose perforated plates and is divided into three compartments, in which is placed the filtering material, by vertical division plates. These partitions are so arranged that the water in passing through the filter will flow under and over in succession, thus assuring the filtering material being always submerged.

Each tank is provided with the following principal connections:

1 6-inch main hotwell pump discharge;
1 3-inch dynamo condenser hotwell pump discharge;
1 4-inch reserve-feed pump discharge;

1 10-inch cross-connecting pipe;
1 8-inch overflow;
steam drain discharges;
vapor pipes.

Feed-Water Heaters.—There is a Reilly multicoil feed-water heater in each fireroom.

Each heater has 239.85 square feet of heating surface, and a rated capacity of 166,700 pounds of water per hour from 100 degrees F. to 200 degrees F., with steam at 10 pounds gage.

The heating agent is the exhaust steam, a back pressure of about 10 pounds gage being kept in the auxiliary exhaust line for this purpose by means of a spring-relief valve at each connection to a condenser, opening toward the condensers.

Main Feed Pumps.—There are three horizontal, direct-connected, turbine-driven, centrifugal main feed pumps located in the center engine room. The speed of the pumps is controlled by "Ideal" pressure governors. Centrifugal, emergency-trip, over-speed, governors are also provided.

The following is the principal pump and turbine data:

Pump—Make and type, Cameron, horizontal, 4-stage.
Capacity, each, 500 G.P.M.
Head, 900 feet.

Turbine—Make and type, G. E. Co., horizontal, single stage.
Horsepower each, 190.
R.p.m., 3,600.

Auxiliary Feed Pumps.—An auxiliary feed pump is installed in each of the three firerooms. They are of the Blake, 16-inch by 10½-inch by 24-inch, vertical, piston, double-acting, single type. "Foster" pressure governors are provided for the speed control of these pumps.

Reserve Feed Pump.—A Blake, 7-inch by 8-inch by 12-inch, vertical, piston, double-acting, single, reserve feed pump is installed in the center engine room. It is arranged to pump water from the reserve-feed tanks to the main feed tanks. It also has a suction connection from the hotwell-pumps cross-connection.

Piping Arrangement.—The feed piping is seamless-drawn copper.

Each main feed pump has a 6-inch independent suction from the main feed tank cross-connecting pipe.

The main feed pumps discharge via the feed-water heaters, or by-pass same if desired, to the boilers. The discharge from each pump is 5 inches, uniting in the center engine room into an 8½-inch connection, leading forward and diminishing in size as it advances to 5 inches in the forward fireroom. The connections to the feed-water heaters are 5 inches in diameter each, and the supply branches to each boiler are 3½ inches in diameter.

The auxiliary-feed suction main is taken off the feed tank's cross-connecting pipe. This main is 10 inches in diameter, reducing in size as it leads forward to the auxiliary feed pumps. Each pump has a 6-inch suction connection.

Each auxiliary feed pump is arranged to discharge to the boilers in its compartment, through the feed-water heater or direct. They can also discharge into the main feed-discharge line to the boilers in other compartments. The discharge connection from each pump is 5 inches in diameter, and the supply branches to the boilers are 3 inches diameter each.

INTERIOR COMMUNICATION.

The customary engine and fireroom telegraphs, gongs, telephones, voice tubes, etc., are fitted for transmitting orders and signaling to the various machinery compartments and other parts of the vessel.

AIR COMPRESSOR PLANT.

An air compressing plant is provided for ejecting gas from the guns, cleaning boiler tubes, cleaning dynamos, running pneumatic tools, etc.

The plant is divided into two parts, each consisting of two motor-driven, geared, compressors and three air-storage tanks,

together with switchboards and appurtenances. One group of equipment is located forward on the third deck in the windlass room, and the other is located on the third deck aft of No. 3 barbette and enclosed in a wire-mesh bulkhead.

The air compressors are automatically controlled so as to maintain a constant pressure on the system. They are also provided with automatic unloading valves so that after shutting down the compressor will always be started up in the unloaded condition.

There is a pneumatic main running through the machinery space with nozzles in each compartment. Branches from this main are led to the general workshop, dynamo, ice-machine and evaporator rooms, galleys, etc., as required. The main is connected to all the air compressors.

The main for ejecting gas from the guns is independent of the above mentioned pneumatic main, and is likewise connected to all the air compressors.

The following are the principal characteristics of the air compressors:

Make—Worthington, motor-driven, geared.

Size—15-inch by 12¾-inch by 7-inch stroke.

Capacity, each—175 cubic feet of free air per minute at 150 pounds' pressure per square inch.

R.p.m., compressor, 300.

R.p.m., motor, 1,600.

Horsepower of motor, 45.

Capacity air-storage tanks, each, cubic feet, 34.

EVAPORATING AND DISTILLING APPARATUS.

The evaporator room is on the second platform between the engine and firerooms, starboard side. The plant is arranged to operate in double-effect with live steam at 70 pounds gage in the first-effect steam coils, or in single-effect when using auxiliary exhaust steam at about 10 pounds gage. Under the latter condition a slight vacuum is maintained on the distillers, an air pump being provided for the purpose.

The plant has a combined nominal capacity of 40,200 gallons of water per 24 hours, when operating in double effect, with a 40 per cent. overload capacity when clean.

EVAPORATOR DATA.

Type	Schutte-Koerting, vertical U-tubes.
Number	6
Diameter, inside, inches	38⅛
Height over all, feet and inches	7–6
Tubes, number (each)	54
diameter of pipe, inches	01¼
thickness of pipe, inch	0.083
Heating surface, square feet (each)	113
Diameter of steam connection, inches	02½
vapor nozzle, inches	07
feed valve, inches	01¼
blow valve, inches	02½

DISTILLER DATA.

Type	Bureau, straight tube.
Number	4
Diameter, inside, inches	22
Length, over all, feet and inches	6–5
Tubes, number (each)	327
diameter, inch	00⅝
thickness, inch	00.065
Cooling surface, square feet (each)	240
Diameter of circulating water inlet and outlet, inches	06
vapor inlet, inches	07
drain, inches	03

EVAPORATOR VAPOR FEED WATER HEATER DATA.

Type	Bureau, U-tube.
Number	2
Tubes, number (each)	12
diameter, inside inches	02½
thickness, inch	0.083
Heating surface, square feet (each)	24.9
Diameter of feed inlet and outlet, inches	02
vapor inlet and outlet inches	07

EVAPORATOR COIL DRAIN FEED WATER HEATER DATA.

Type	New York Navy Yard, U-tube.
Number	2
Tubes, number (each)	6
diameter (I.P.S.), inches	1

Tubes, thickness, inch	0.134
Heating surface, square feet (each)	12.5
Diameter of feed inlet and outlet, inches	1¼
coil drain inlet and outlet, inches	1½

PUMP DATA.

Evaporator feed pumps:
Number and type—(2)—Worthington, vertical, piston, double-acting, single.
Size....4½-inch by 5-inch by 6-inch.
Evaporator brine pump:
Type....Worthington, vertical, piston, double-acting, single.
Size....4½-inch by 5-inch by 6-inch.
Distiller circulating pumps:
Number and type—(2)—Worthington, horizontal, turbine-driven, centrifugal.
Capacity (each), G.p.m. at 25 feet head.... 800
Turbine, type ....Terry.
R.p.m. 2,000
Horsepower (each) 12
Distiller fresh-water pumps:
Number and type—(2)—Worthington, vertical, piston, double-acting, single.
Size....4½-inch by 5-inch by 6-inch.
Distiller air pump:
Type—Worthington, vertical, piston, double-acting, single, featherweight.
Size....6-inch by 10-inch by 8-inch.

The quantity of fresh water produced by the plant is measured by a Cochrane precision meter.

GENERAL WORKSHOP.

The following machine tools, each driven by its own electric motor, are installed in the machine shop on the third deck, over the center engine room:

No.	Description.	Make and H.P. of motor.
1	28-inch by 48-inch swing, extension-gap lathe.	
1	16½-inch swing lathe; American Tool Works	Reliance, 3 H.P.
1	14-inch swing lathe; American Tool Works.	Reliance, 3 H.P.

1 8-inch swing precision lathe.
1 24-inch column shaper; American Tool Works ....Reliance, 3 H.P.
1 30-inch radial drill; American Tool Works. Reliance, 3 H.P.
1 16-inch sensitive drill, Sipp Machine Co... ¾ H. P.
1 Universal milling machine; Kearney & Trecker ....Reliance, 3 H.P.
1 Floor grinder; Willey..................Reliance, 2 H.P.
2 Portable cylinder-boring machines;
1 Portable surface grinder; Cincinnati Electric Tool Co........................ 1 H.P.
1 4-inch hacksaw.

BLACKSMITH SHOP.

A blacksmith shop is provided on the superstructure deck, in the foundry enclosure. It is equipped with one portable and one permanent forge, together with anvil and all necessary tools and fittings. The permanent forge is oil burning; it is fitted with an electrically-driven blast fan. The portable forge is of the coal-burning folding type.

FOUNDRY.

A small foundry is installed on the superstructure deck. The outfit consists of a small marine, oil-burning, tilting, crucible furnace, together with adequate allowance of crucibles and the necessary apparatus for handling and pouring the metal.

PIPE AND COPPERSMITH SHOP.

The pipe and coppersmiths' equipment is located on the third deck in the forward boiler hatch, and consists of the following:

1 Motor-driven, pipe-cutting and threading machine; 2-inch to 6-inch capacity;
1 tinner's combined punch and shears;
1 tinner's slip roll forming machine;

1 tinner's beading machine;
1 tinner's cornice brake;
1 oxyacetylene welding outfit;
1 oil forge;
vises, work-benches, etc.

ELECTRIC PLANT.

There are two dynamo rooms, one just forward of the forward boiler room and the other just aft of the after boiler room. They are on the second platform level.

The distribution rooms, two in number, are located on the first platform, one over each dynamo room. They contain the lighting, power, searchlight and generator boards combined.

The generator installation consists of four horizontal, 6-pole, compound-wound, 300-kilowatt, 3-wire, General Electric generators, two in each dynamo room; they are gear-driven by three-stage horizontal Curtis turbines, designed to run at 5,037 revolutions per minute, with steam at 200 pounds gage. Each generator will deliver at normal load 1,250 ampères of current at 240 volts, when running at 1,000 revolutions per minute. The generators are capable of delivering one-third overload for two hours without injury.

Steam for the generators is taken off the main steam cross-connection in the forward and after firerooms, for the forward and after generators, respectively. The branch to each genertor is 4 inches and fitted with stop, reducing and throttle valves.

The generators are arranged with condensing and atmospheric exhaust, the latter connections, via escape pipe, are 10 inches and fitted with automatic valves permitting instantaneous change over in emergency.

Each pair of generators has an independent condensing apparatus, located below the generators in a separate room in the hold, with access hatch between the two levels. Each condensing apparatus consists of one condenser and vacuum

augmenter, together with independent air pump, circulating pump, hotwell-tank pump and hotwell tank of 300 gallons' capacity.

The dynamo condensers also handle the exhaust in port.

DYNAMO CONDENSER DATA.

Number	2
Inside diameter, feet and inches	4–05
Thickness of shell (steel), inch	0⅜
Length between tube sheets, feet and inches	8–6½
Thickness of tube sheets, inch	01⅛
Tubes, number (each)	1,112
diameter, outside, inch	00⅝
thickness, inch	00.065
Cooling surface, square feet (each)	2,006
Diameter of exhaust nozzles, generators (2), inches	18
augmenter suction, inches	05
air-pump suctions, inches	07
circulating-water inlet and outlet, inches	09

AUGMENTER CONDENSER DATA.

Number	2
Inside diameter, inches	12½
Thickness of shell (composition), inch	0⅜
Length between tube sheets, inches	32
Thickness of tube sheets, inch	01
Tubes, number (each)	116
diameter, outside, inch	00⅝
thickness, inch	00.049
Cooling surface, square feet (each)	50.61
Diameter vapor inlet, inches	04
air-pump suction, inches	04
circulating-water inlet and outlet, inches	03

PUMP DATA.

Dynamo condenser circulating pumps:

Number and type—(2)—Worthington, horizontal, turbine-driven, centrifugal.

Capacity (each), G.p.m. at 20 feet head	2,000
Turbine, type	Terry.
R.p.m.	1,300
Horsepower (each)	15.7

Dynamo condenser air pumps:

Number and type—(2)—Worthington, vertical, twin, beam, single-acting.

Size.............................9-inch by 18-inch by 18-inch by 12-inch.

Dynamo condenser hotwell pumps:

Number and type—(2)—Worthington, vertical, piston, double-acting, single.

Size...................................4½-inch by 5-inch by 6-inch.

TRIALS.

The *New Mexico* has had two sets of official trials. The first trials were conducted after the ship had been out of dry dock about seven weeks, and this increased the shaft horsepower necessary to make the guaranteed speed of 21 knots by over 4,000 horsepower, so the ship was docked, cleaned and painted, and a second set of trials were run when the horsepower came down to 29,100 for 21 knots. The ship averaged 21.08 knots for her four-hour full-power run and 21.31 knots was the average of the five high points on standardization.

The turbine water rates per shaft horsepower were considerably better on the first trials than on the second, owing to the fact that the first trials were run in smooth water, while the second trials were run in rough weather, which increased the horsepower for the speed over that obtained on the standardization runs and for which no allowance was made, as the standardization horsepower was used. The water-rates obtained on the trials were as follows:

Speed.	Water rate first trial.	Water rate second trial.
Full speed	12.29	12.01
19 Knots	11.926	12.33
15 Knots..............	11.667	12.475
10 Knots	14.223	13.96

These water rates include the steam from the 300-kw. exciters, which furnished power for excitation and also for driving the main air pumps, main circulating pumps, main condensate pumps, forced-lubrication pumps, and ventilating blowers for the main motors.

In addition to the trials for steam and fuel consumption, trials were made to demonstrate satisfactory backing qualities when steaming at full power; also to determine satisfactory performance when turning with full rudder and at full speed, both in the case of propulsion by one generator and also by two generators. Numerous other trials were conducted, such as standardizing with two screws driving and two running idle, and considerable new data was obtained in regard to the action of screw propellers. In all the tests carried out the machinery gave very satisfactory performance.

The average trial data obtained on the second trial is given in the following table:

	4-hour full power trial.	4-hour 19-knot trial.	4-hour 15-knot trial.	1-hour 10-knot trial.
Steam at boilers, lbs. gage	278.6	274.8	273.8	....
turbines, lbs. gage	272.1	274.2	274.6	277.5
turb. 1st stage, lbs. gage.	139.7	104.4	86.4	42.5
Vacuum, inches	29	29.5	29.8	30.
Barometer, inches	30.83	30.77	30.92	30.79
Fireroom air pressure inches of water	4.1	3.4	2.3	1.7
Feed water temperature, deg. F.	182.8	189.7	203.4	208.9
Main generators, volts	4257.	3740.	2915.6	1950.
ampères	1873.5	1565.	2206.2	1600.
field, volts	171.7	152.15	143.8	118.
field, ampères	318.25	285.6	290.	245.
r.p.m.	2042.	1825.5	2012.5	1450.
Main motors, ampères	994.5	860.3	572.5	417.5
r.p.m.	167.69	152.2	115.35	80.49
Slip of propellers, per cent	16.	14.97	13.24	14.83
Speed in knots	21.08	19.37	14.98	10.26
S.H.P. (from curve)	31197.	23233.	9648.	3690.

BALANCING MACHINES.

F. G. HECHLER, ASSOCIATE.*

The various articles on balancing which have recently appeared in this and other journals attest to the increasing interest manifested in this very important subject. The use of high speed rotative machinery has brought the matter prominently before the marine engineer, and he sees in it a possible solution for some of the baffling problems of excessive vibration that confront him. A little reflection will make it clear that the supports for machinery on shipboard are in general less rigid than in stationary work, and that the possibility of setting up a vibration of the ship's frame which will be in synchronism with that of a rotating machine, is by no means remote. It is not unusual to find that at certain speeds, which must be maintained when cruising in squadron formation, the vibration is very trying on the personnel. This phenomenon of vibrational "Critical Speed," as it is often called, is in many cases probably due to synchronism between an unbalanced turbine rotor or propeller, for instance, and the hull. In addition to the discomfort of the crew, the unbalance causes undue wear of the bearings, commutation and lubrication troubles and a more or less serious loss of power.

Although the lack of proper balance is quite generally recognized, there is considerable confusion regarding the subject and especially as to the methods to be used in securing both static and dynamic balance. Only recently a midshipman of the first class, who in a few months will become an officer, said to the writer that he had been told that the subject was so difficult that only two or three people knew the formulae therefor, and that they could get almost any price for their services! It is to be hoped that such ideas of the subject may soon be dispelled.

*Mechanical Engineer, U. S. Naval Engineering Experiment Station, Annapolis, Md.

In the past, the main reliance has been placed on securing satisfactory static balance by using the ordinary parallel ways, or sometimes roller or ball bearings. Except under the most favorable conditions of light weight and small shaft, combined with a relatively large radius of gyration, the static balance thus obtained will not be very exact as the method is not sufficiently sensitive. However, if after the static balance is satisfactorily adjusted, and the machine runs smoothly at all speeds, well and good; if not, an effort is made, usually by trial and error, to improve conditions by adding weights here and there until by chance the running becomes satisfactory. At best the method is slow and uncertain.

Occasionally, when the importance of the subject was more fully appreciated, balancing machines were designed and built. These machines have usually been called " dynamic balancing machines," but have in reality been a rather curious mixture of both static and dynamic, in that no provision was made for properly separating the two effects. Unless a body is first put in perfect static balance it is useless to attempt to balance it dynamically, for the results will surely be misleading and may be disastrous. These machines have depended usually either on the use of a floating bearing at each end, with all motion suppressed except in the horizontal plane, or on the one-point-suspension principle, as for instance, when a flywheel, pivoted on a cone near its center of gravity, is rotated in a horizontal plane. In both cases, the body runs out and the " heavy spot" is marked in some way. Usually the body is rotated in both directions, giving two distinct sets of marks from which the plane of unbalance must be determined. Then too, strange though it may seem, the effort was often made to correct what was assumed to be a dynamic unbalance by adding or removing weight at one point only! It can readily be shown that neither the floating bearing nor the one-point-suspension method is based on sound theoretical grounds, and hence machines built in accordance therewith are not, as a rule, susceptible of satisfactory use. However, in the hands of

capable and experienced men some very good operating results have apparently been obtained, but not with the definite directness and ease that is possible.

Credit for developing and applying the correct methods based on a rational treatment of the subject is due Mr. N. W Akimoff. In a paper presented at the New Orleans meeting of the American Society of Mechanical Engineers in April, 1916, Mr. Akimoff first presented the subject in a most interesting and readable manner. Since then, however, great progress has been made in the application of the fundamental principles there laid down.

The subject is of great practical importance, and a description of the design and method of operation of the two leading makes of balancing machines now in the field will be of interest. Both the balancing machines to be described have been under test at the U. S. Naval Engineering Experiment Station, Annapolis, Maryland, within the past year.

Before describing the machines, however, it is desirable to examine briefly the meaning of static and dynamic balance. Fig. 1 represents a body symmetrical about the axis, but which

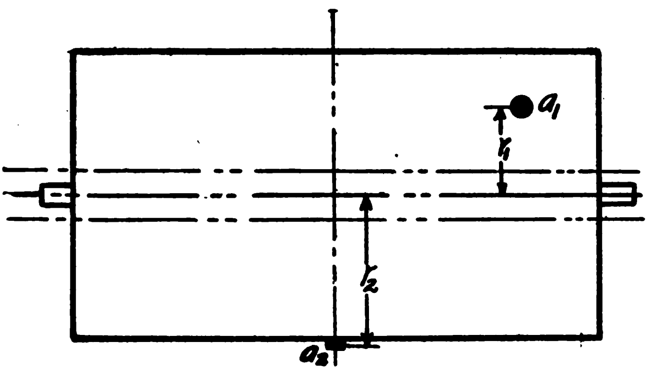


FIG. 1.

is out of static balance because of the "heavy spot" a_1 at a radius r_1. If such a body is placed on sensitive parallel ways it will always come to rest with a_1 vertically below the axis.

When the body is rotated, the unbalanced force due to a_1 will tend to make the axis "throw out," as indicated by the dotted center lines; that is, the body suffers a translation. To establish static balance it is only necessary to add one weight as a_2 at radius r_2 in the same axial plane as a_1, and such that $a_2 r_2 = a_1 r_1$. In general the "heavy" spot is not due to a single disturbing element, but represents the resultant effect of all unbalanced elements, and hence static balance for the entire body may be obtained by adding or removing a single weight. Evidently a body is in static balance when its center of gravity lies on the axis of rotation.

Dynamic unbalance in a statically balanced body is caused by a centrifugal couple in some axial plane. The centrifugal couple tends to produce rotation about an axis through the center of gravity of the body as shown by the arrows in Fig. 2.

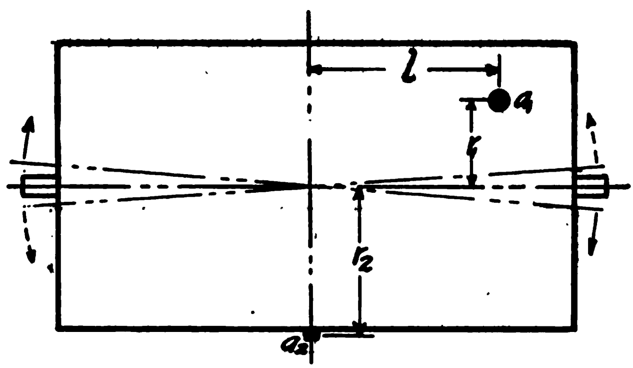


FIG. 2.

Static balance may be obtained by adding a single weight, but there is only one position in which the weight may be put, namely, directly opposite the resultant of the unbalanced static forces, without introducing a new centrifugal couple. This couple when combined with those that may already be present may give a resultant centrifugal couple either greater or smaller depending on the conditions. In the floating-bearing

type of machine the body is rotated and the high spot marked. The mark does not, however, indicate the plane of the disturbing couple; neither is its plane just 90 degrees ahead of or behind the mark obtained. The angle between the two is really a very elusive thing depending among other things on air resistance and the angular velocity of rotation of the body, and had, therefore, best be avoided altogether.

UNITS FOR STATIC AND DYNAMIC UNBALANCE.

Since static balance is obtained by adding a weight whose moment about the axis is equal to that of the disturbing element, it must be evident that the most satisfactory method of expressing the amount of static unbalance is in ounce-inches, that is, as the product of the weight necessary to restore balance multiplied by the radial distance at which it is applied.

Dynamic unbalance is caused by two centrifugal forces each equal, since $a_1 r_1 = a_2 r_2$, to $\omega^2 a_1 r_1/g$, where ω is the angular velocity of the rotating body and g is the acceleration due to gravity. If, as shown in Fig. 2, these two forces act on opposite sides of the axis at an axial distance l apart, they produce a centrifugal couple equal to $(\omega^2 a_1 r_1/g)\ l$, which may be written as $K\ a_1 r_1 l$. That is dynamic unbalance is equal to some factor K times weight times radial distance times axial distance. The factor K is a variable depending on the angular velocity, but its value must be the same for the disturbing and the neutralizing couple because both necessarily rotate at the same speed; therefore, for convenience, and because it would be difficult to determine its actual value, we may agree to disregard K, whence the dynamic unbalance is proportional to $r_1 a_1 l$, or its unit of measurement becomes inch-ounce inches. Each of the terms of this expression is easily measured, and the result is, therefore, free from all uncertainty that would be involved by retaining the term K.

REQUIREMENTS FOR A SATISFACTORY BALANCING MACHINE.

Until recently there was no machine with which static and

dynamic unbalance could be differentiated. It was assumed that static balance could be satisfactorily obtained by using horizontal balancing ways or their equivalent. Now the quality of the dynamic balance will be good or bad according as the static balancing has been well or poorly done; it is therefore absolutely necessary that the static balance should be perfect before attempting to obtain dynamic balance. Hence the ideal balancing machine should fulfill the following five requirements:

(1) Provide a sensitive and accurate method for determining static balance.

(2) Provide an efficient method for finding dynamic balance.

(3) Indicate the magnitude of the unbalanced force or couple.

(4) Indicate the axial plane of the unbalanced force or couple.

(5) Indicate the sign or direction of the unbalanced force or couple.

So far as the writer is aware, there are only two machines which adequately meet any of these requirements; namely, the Carwen Dynamic Balancing Machine, manufactured by the Carlson Wenstrom Company, and the Combined Static and Dynamic Balancing Machine, built by the Vibration Specialty Company.

CARWEN DYNAMIC BALANCING MACHINE.

The Carwen machine is built in nine sizes having capacities varying from 3/4 pounds to 14,000 pounds and capable of handling objects up to 14 feet long and 60 inches in diameter. One of the intermediate sizes especially adapted for handling automobile crankshafts and small armatures, is shown in the photograph, Fig. 3. Two substantial box pedestals, one at either end of the machine, are rigidly connected by two tie

rods one on either side. The pedestals support a frame similar to a lathe bed. At one end the bed is hinged to the pedestal by a flat plate spring while the other end rests on helical springs. This arrangement allows the free end of the bed to vibrate up and down in a vertical plane and at the same time prevents motion in all other directions. Suspended underneath the vibrating frame is a small electric motor and the two counter balancing weights marked D, D. These weights are splined to the supporting shaft so that a pair of forked arms moved by a right and left screw geared to the hand wheel A, may move them away from or toward each other. When the weights are together they form a symmetrical body in perfect balance; when they are separated a compensating centrifugal couple is introduced.

The body to be balanced is mounted either on adjustable pedestal bearings attached to the bed, or, if it is a chankshaft, preferably in its own base so that it will be supported the same as in actual use. A balanced driving head is fastened to the body, and by means of flexible steel connectors or clips the body and the compensating weights are driven at exactly the same speed and in the same direction by the motor attached to the bed. The shaft on which the weights are placed is driven through a planetary gear to permit the plane of the compensating weights to be brought into the same plane as the disturbing couple in the body. By turning the hand wheel C the angular position of the gear and weights may be altered without stopping the machine.

In using this machine it is absolutely necessary that the object shall be in as nearly perfect static balance as it is possible to obtain by any available method. Neglect of this precaution may lead to erroneous and unsatisfactory results for which the operator has only himself to blame. It must therefore, be assumed that the body is in perfect static balance and that the reading of the machine represents pure dynamic unbalance only.

When the motor is started and the speed is gradually increased, a point will soon be found where the bed has maximum uniform vibrations as shown by the indicator M, which is an ordinary dial indicator graduated in thousandths of an inch. This is the synchronous speed at which the natural period of the springs and the bed is the same as the forced period due to the rotating body. By changing the number of springs, or their characteristics, it is always possible to have synchronism come at a comparatively low speed, say less than 500 revolutions per minute. If the speed is either too high or too low the vibrations will be irregular, periodically increasing and decreasing in amplitude, and perhaps momentarily stopping altogether. By changing the angular position of the compensating weights and their axial distance apart, the position is readily found in which the dynamic unbalance of the body is neutralized.

The value of the disturbing couple in inch-ounce-inches is indicated by a pointer on the dial B. Its plane will coincide with that of the compensating weights, and may therefore be readily found; its sign or direction will be opposite to that of the weights. The machine therefore gives positive and exact information regarding the dynamic unbalance provided the static balance was perfect. It meets all but the first of the five requirements for an ideal balancing machine. It is sensitive and accurate and results are readily obtained, as the adjustments are quickly made without stopping the machine. The compensating weights built integral with the machine restrict its capacity; the machine illustrated has a maximum capacity for neutralizing dynamic unbalance of only 288 inch-ounce-inches. To increase the range by using heavier weights would make it less sensitive.

VIBRATION SPECIALTY COMPANY'S BALANCING MACHINE.

The appearance of the Vibration Specialty Company's Combined Static and Dynamic Balancing Machine is shown in Fig. 4. It will be noted that the unit is relatively small and

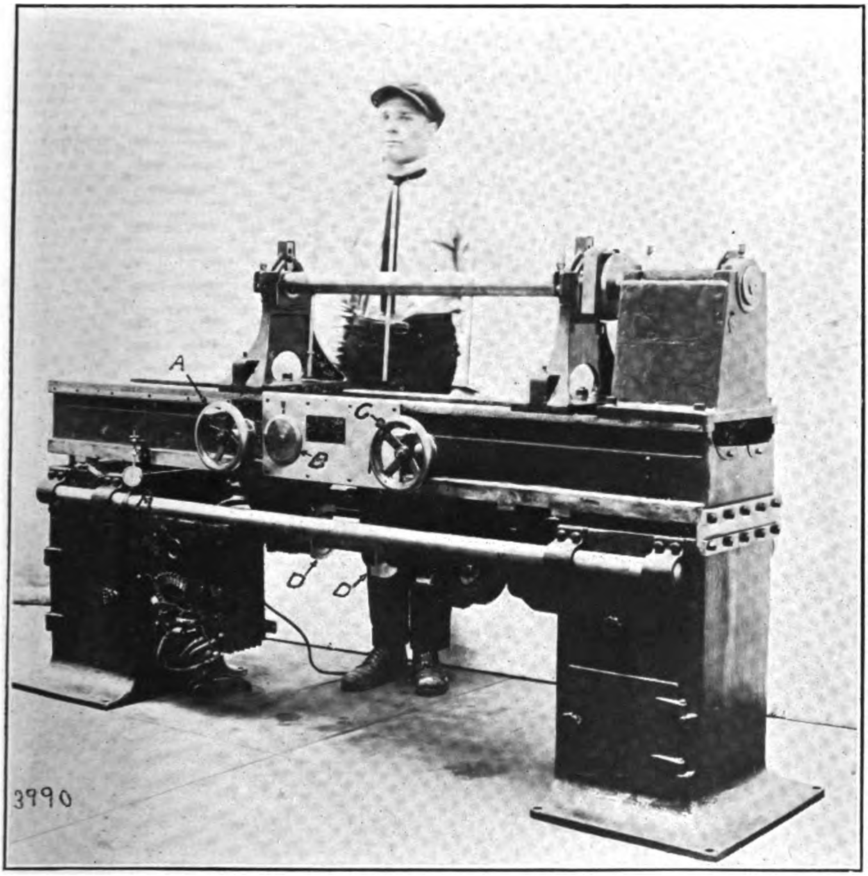


FIG. 3.

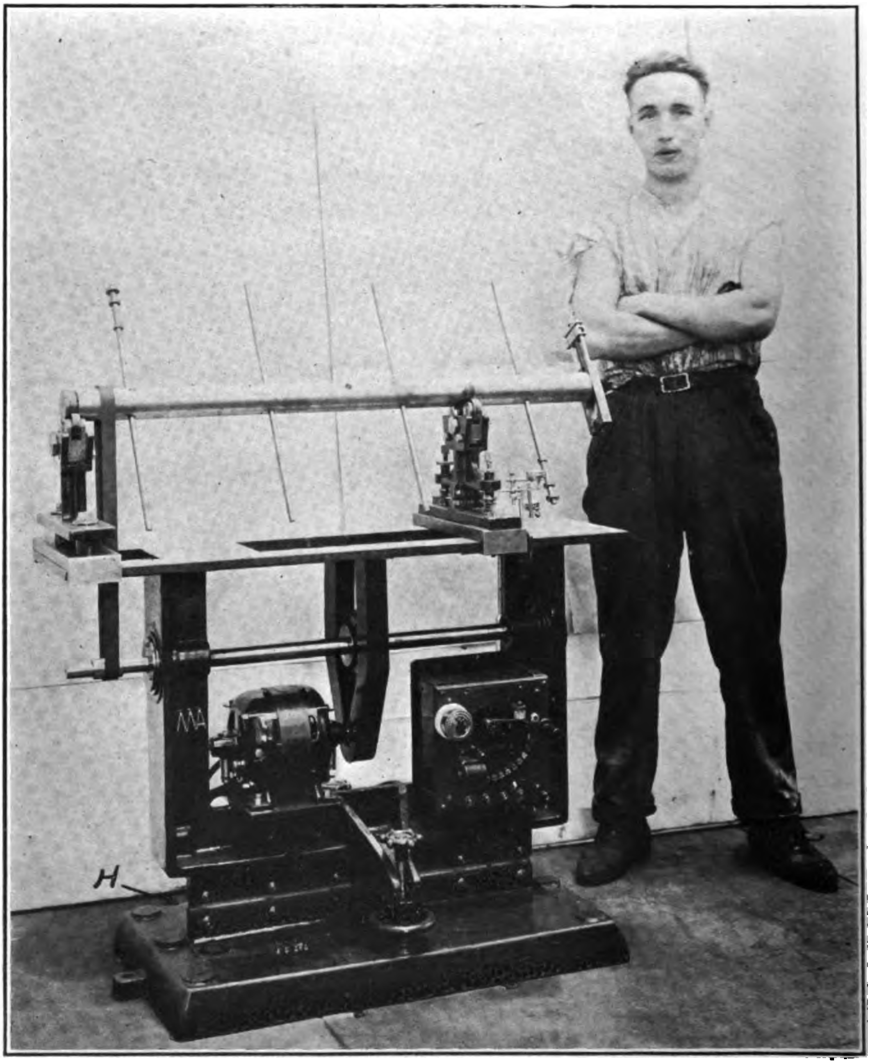


Fig. 4.

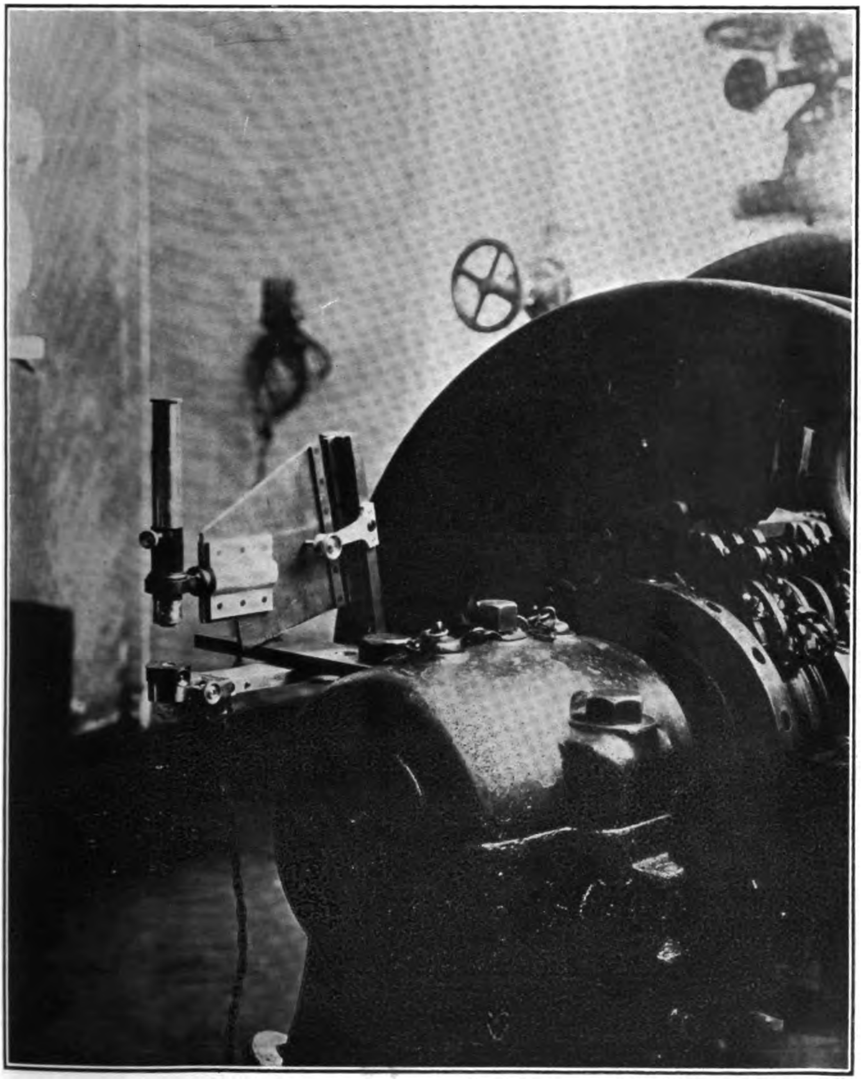


FIG. 5.

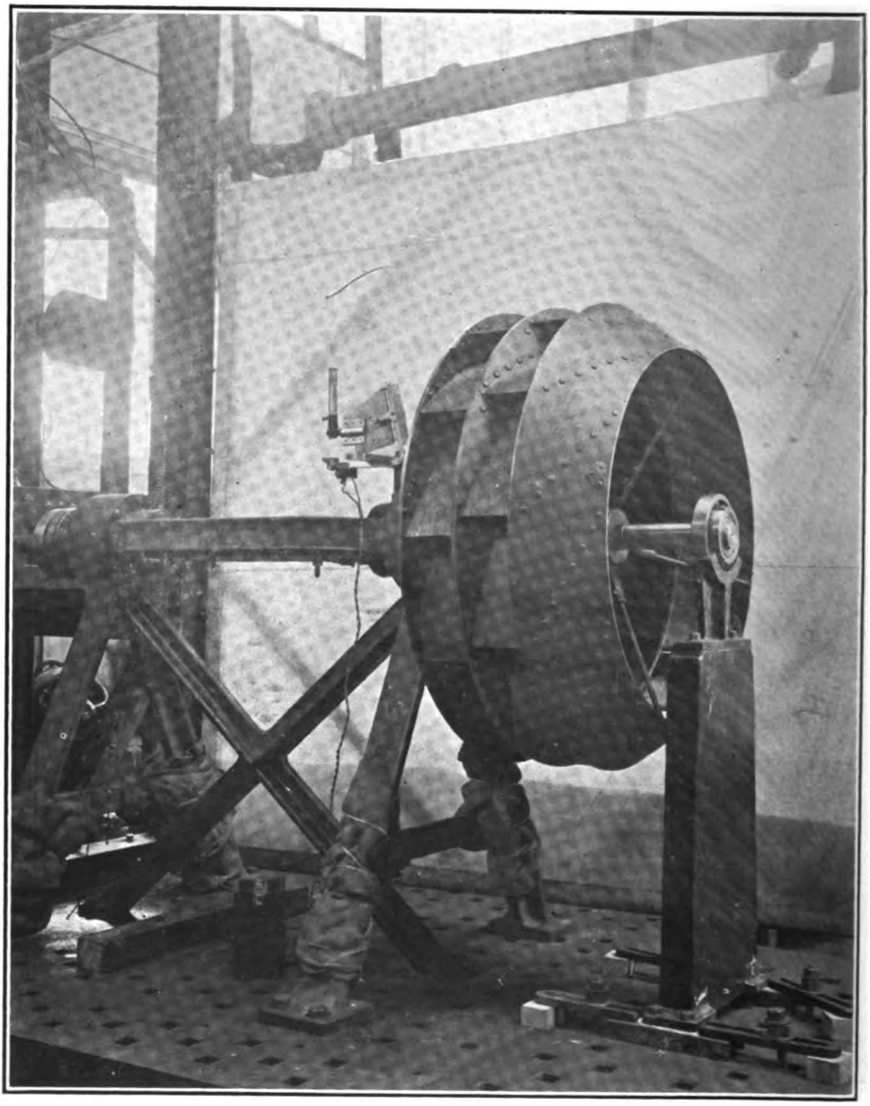


FIG. 6.

compact and simple in construction. A cast-iron box base supports a vertical frame by means of a steel-plate hinge H which permits the frame to vibrate horizontally in one direction only. An arm extends out from each side of the frame and carries at its end an adjusting screw and a hand wheel. By means of the hand wheel the tension of a helical spring beneath each arm may be adjusted in order to increase or decrease the natural period of vibration of the frame. The adjusting screws are also used for rigidly locking the frame when the machine is used for dynamic balancing. The frame carries a small motor with its starting and speed control rheostat. The countershaft just above the motor is driven by a chain drive. A rubber belt at one end of the counter shaft is used to drive the object to be balanced.

The frame carries two pedestal supports on which the object to be balanced is mounted on rollers. The left-hand pedestal is rigid, thus differing from the right-hand one, which is supported on a thin phosphor-bronze plate hinge in the same manner as the frame is attached to the base, and which in the same way permits vibration of the pedestal in one direction only. An arm also extends out on each side of the pedestal, with adjusting screws and small helical springs for changing the natural period of vibration of the pedestal, or for locking it rigidly in position when determining static balance.

Two dial indicators are shown at the right-hand end of the machine. One of the indicators is supported by the frame and shows the amount of vibration of the pedestal; the other is supported from the base of the machine and indicates the vibration of the frame itself.

It will be observed that no compensating couple is used in this machine; instead a balancing clamp is attached to one end of the object to be balanced, and hence, rotates with it. The balancing clamp is shown in position on the test arbor in Fig. 4. The construction of the clamp is very simple; it may readily be fastened in any angular position and its neutralizing effect changed by moving the sliding weight nearer to or far-

ther from the center. As shown the weight is nearly all the way out, where it has its maximum compensating effect, and the clamp is in the proper plane to neutralize the centrifugal couple due to the small weights on the ends of the two arms of the test arbor. In using this machine the first step is to check the static balance of the body under test. For this purpose the pedestal support on the frame is rigidly locked and the frame is released from the base so that it may vibrate freely due to any unbalanced static force which will tend to cause a translation of the body, as in Fig. 1. The motor is then started and the body rotated, the speed is varied, and, if necessary, the tension of the springs changed, until the vibration as shown by the indicator becomes a maximum. The clamp is then applied and its position and that of the sliding weight shifted until the resultant of the unbalanced static force is neutralized and the vibration reduced to zero. The position of the clamp then shows the plane of the resultant of the unbalanced static forces while its value in ounce-inches is given by the weight in ounces of the sliding weight multiplied by the distance it has been moved from its neutral position on the clamp. The correction is made by adding or removing weight at one point; preferably, in the transverse plane through the center of gravity of the body.

After the body is perfectly balanced statically, the frame is rigidly locked to the base by the arms and adjusting screws, and the right-hand pedestal is unclamped. The body is again rotated at various speeds until the synchronous speed, at which the vibrations are a maximum and uniform, is found. The balancing clamp is then applied and the positions of the clamp and of the movable weight are shifted until the dynamic unbalance is neutralized and the vibration becomes zero. The unbalanced centrifugal couple is then numerically equal to the weight of the movable weight multiplied by the radial distance it was moved from its neutral position multiplied by the axial distance from the central plane of the clamp to the center of the rollers on the left-hand rigid pedestal. The position of

the clamp shows the plane and the sign of the disturbing couple, which may then be neutralized by adding or removing weight at two points in the same axial plane on opposite sides of the axis.

The Vibration Specialty Company's Machine meets all five of the requirements and does so in a really simple manner, much less complicated than might be supposed from the somewhat involved description just given.

It is, of course, rather inconvenient to stop the machine each time the position of the clamp or the neutralizing weight is to be changed, and also to obtain the amount of the unbalance by measurement and computation. An experienced operator, however, usually does not have much difficulty in quickly obtaining a satisfactory result. On the other hand, the balancing clamp possesses certain valuable features which in the writer's opinion more than outweigh any disadvantages it may have. If the body is badly unbalanced so that a single clamp is insufficient, two or more may be used; moreover, these need not be of the same size, thus one may have a small movable weight, or even two weights, one large and the other small, so that the final adjustment may be accurately made with the small weight.

The machine is exceptionally sensitive and accurate for both static and dynamic balancing. For static balancing in particular the machine is far more sensitive than balancing ways; this is of great importance because good dynamic balance cannot be obtained unless the static balance is first good. Dynamic unbalance has no effect on static balance, but the reverse is not true.

When arranged as in Fig. 4 the machine is adapted to handle objects up to 20 inches in diameter, but by the addition of certain auxiliary attachments it is possible to use the same machine for objects up to 60 inches in diameter and weighing, say, up to 1,000 pounds. With such large objects it is necessary to use a large balancing clamp. Hence, by using the auxiliary attachments and providing a number of balancing

clamps of different sizes the machine may be adapted to handle a large variety of work. Turbine rotors weighing up to sixteen tons have been successfully balanced on larger machines built by the Vibration Specialty Company.

LIMITS OF BALANCE.

With satisfactory balancing machines available, with one of which at least it is possible to distinguish between static and dynamic unbalance, it seems desirable that certain limits or tolerances for unbalance should be established. These limits should be included in the specifications for the various types of machinery. Just what they should be probably no one now knows, and they can be fixed only by experience and a thorough investigation of the entire subject. Undoubtedly, the limits will vary, not only for various classes of machinery, but also with the size and weight. Mr. Akimoff has proposed tentatively that for bodies weighing not over 150 pounds and with journal diameters of not over three inches the limit for static balance should be 0.4 ounce-inches, with a corresponding limit for dynamic balance fixed at 12 inch-ounce-inches.

AMPLITUDE OF VIBRATION IN UNBALANCED BODIES.

Obviously if a rotor is perfectly balanced it will set up no vibrations. If the balance is not perfect there will be more or less vibration, and sometimes it is difficult to determine whether or not the amount of vibration is excessive. Often an inspector is in doubt whether to pass or reject a machine because it vibrates at certain speeds. He knows that the manufacturer balanced it as well as he could without using a balancing machine constructed on a sound theoretical basis, and besides he has no way to determine the actual amplitude of the vibration, he can only judge that it is good or bad by the "feel." Or perhaps, it is necessary to improve the balance of a unit already installed; weights are added here and

there, the unit is brought up to speed, and quite likely a discussion ensues between different observers as to whether the balance is better or worse than before.

To answer these perplexing questions the Vibration Specialty Company has recently developed an instrument called a Vibroscope. The first one built was recently tested at the Engineering Experiment Station with interesting results. The Vibroscope is shown in Fig. 5, attached to the outboard bearing of a horizontal turbo generator. It is built on the principle of the seismograph; the frame or vibrating part of the instrument is arranged so that it may be conveniently attached to the machine whose amplitude of vibration is to be determined. The upper leg of the frame is slightly inclined and has attached to it by a thin, flexible spring hinge a heavy, flat-plate pendulum supporting a micrometer microscope at its outer end. This pendulum plate has a very slow period and becomes the "steady point" of the arrangement. Directly beneath the microscope at the end of the horizontal part of the frame is a narrow slit illuminated by an electric light. By looking through the micrometer microscope with the instrument at rest the width of the slot is adjusted to, say, five divisions on the illuminated scale; then with the machine running any apparent increase in width of the light slit is due to vibration. By calibrating the instrument the relation between the reading of the Vibroscope and the amplitude of the vibration may be determined.

To obtain some idea of the vibration of actual machines the Vibroscope was attached to two turbo-generators as follows:

(a) 300-k.w. turbo-generator set. Speed 1,800 r.p.m. This unit operates satisfactorily and with very little vibration. The outboard bearing at the generator end vibrated about 0.0001 inch. The turbine shell in a horizontal diametral plane vibrated 0.0003 inch.

(b) 75-k.w. turbo-generator set. Speed 2,400 r.p.m. This unit runs badly, commutation is poor and vibration excessive, particularly, at the main bearing between generator

and turbine. When attached to the end bearing as shown in Fig. 5, the Vibroscope indicated a vibration of 0.0003 inch. The main bearing, however, had an amplitude of 0.0026 inch. It appears, therefore, that for turbo-generators a vibration of 0.0005 inch is permissible, but that 0.0025 inch is not. Where the limit lies between these two values can be determined only by further tests.

BALANCING A DESTROYER FAN.

A 30-inch destroyer fan was selected for test to determine the amount of static and dynamic unbalance and the decrease in the amplitude of vibration due to the elimination of each. The fan, shown in Fig. 6, had an overall diameter of 44 inches, and weighed 555 pounds. The manufacturer had evidently balanced the fan statically as well as his facilities permitted, as was evidenced by a weight at the periphery of the wheel in the plane through the center of gravity.

The balancing was done on the Vibration Specialty Company's machine shown in Fig. 4, with the auxiliary attachments previously referred to. The static unbalance was found to be 48.4 ounce-inches, which is equivalent to 2.2 ounces at the periphery of the fan; the dynamic unbalance was 1,400 inch-ounce-inches, equivalent to 6.2 ounces at the periphery and in two transverse planes 10.6 inches apart axially.

The balanced fan was then mounted as in Fig. 6, and driven by an electric dynamometer through a flexible coupling. The Vibroscope was attached to the pedestal bearing adjacent to the fan. At 1,425 revolutions per minute, the operating speed of the fan, the amplitude of vibration with the fan balanced was 0.0035 inch. By removing the fan from the shaft it was found that the vibration of the shaft alone was still 0.0035 inch at 1,425 revolutions, due to unbalance in the dynamometer and especially to a heavy unbalanced shaft coupling adjacent to and behind the fan. Hence it is reasonable to conclude that the vibration observed with the balanced fan was not due to

the fan itself. When the two dynamic correction weights were removed, leaving the fan in perfect static balance, the amplitude of the vibration increased to 0.0068 inch. Finally after removing the static weight also, thus restoring the fan to the condition in which it was received from the manufacturer, the vibration became 0.01 inch.

The conclusion is obvious. In this case, the manufacturer had balanced the fan, and, as such things go, it did not seem so very bad. But after securing static and dynamic balance the vibration dropped to one-third of its original value, and would undoubtedly have been even less if the supporting shaft had been in balance. Such an improvement is surely well worth while.

In closing, just a word of warning. Although the importance of securing static and dynamic balance can scarcely be overestimated, it must not be thought that vibration is always due to a lack of one or the other. Other disturbing factors, such as torsional vibrations, may be present and must often be taken into account if the problem is to be fully solved.

BALTIMORE AND SOME EARLY YEARS IN THE ENGINEER CORPS, U. S. N.

By Chief Engineer Fred. G. McKean, Member.

The following notes, in their departure from the usual articles in the Journal of the American Society of Naval Engineers as to solid science and the description of recent attainments, call for some explanation.

It is believed that the History of the Corps should be written before the steam engine is possibly superseded by electricity, aviation, or whatever is in the womb of the future; and that when it is written, all details which will throw light upon its affairs will be useful, varying as they might, from Passed Assistant Engineer Bennett's monumental work, the "Steam Navy of the United States," through various other writings to the personal reminiscences of men some of whom may be about to join the majority.

Two recent articles in the Journal emphasize the advisability of such action. The instructive review "Fifty-Year Retrospect of Naval Marine Engineering," by Rear Admiral C. W. Dyson, U. S. N., showed even those who have some memory, that it is refreshing to take stock once in a while of events which have passed; and the interesting summing up in another article sub-entitled, "A Quarter of a Century of the Journal," by Rear Admiral John R. Edwards, showed the danger of delay, for in the preparation of that article it was found that several important points of fact had already become matters of doubt, and necessitated reference to private diaries and to comparison of sometimes treacherous recollection.

If the foregoing remarks are conceded, it remains to state more specifically that the intention of the present review is to mention briefly a few occurrences and persons known in

"the fifties" of the nineteenth century, but also digressing occasionally to times before, and alluding now and then to the almost incredible advances made since then.

Baltimore for several reasons played an important part in the progress of the Engineer Corps, and, in fact, of the United States. Its proximity to the Capital, the production of Government work by its five best known machine shops, its situation as the headquarters of the whole South and connecting link with the whole North, combined to give it prominence; while historically, the previous aid of western Maryland in casting cannon, the operation of well-known furnaces, a rolling mill which later furnished the armor plates of the first Monitor, Rumsey's experiments in early steam navigation on the upper Potomac, the fame of Baltimore Clippers in peace and war, and its achievements almost as a pioneer in railroading and telegraphing, gave it a traditional and deserved pride.

For the purposes of the present compilation it may be stated that among the machine shops of Baltimore the Vulcan Works, at that time under the management of James Murray and Henry R. Hazlehurst, successors of Watchman and Bratt, not only did much Government work—marine, lighthouse, bridge, railing, etc., but included among its other opportunities candidates for entrance into the Engineer Corps, insomuch that the modest and rather dingy drafting room became the rendezvous of several successive groups of aspirants and grades of naval engineers and some outsiders, who brought new questions and contributed matter for inquiry and general profit. The chief draftsman, Mr. Eliakim T. Robb, was unusually fitted to help candidates, being thoroughly competent, able and always cheerfully willing to impart information. Furthermore, the questions asked and answers given at the examinations were being eagerly collected by industrious candidates, and later became bases for monographs upon the various branches, and in at least one instance used after the war as a lecture in the Engineer Department at Annapolis.

Baltimore, in common with the engineering world, discussed warmly and intelligently the problems of the day; thus, it may surprise some readers that the question of the possible rejection of surface condensation in seagoing steamers, the disbelief of many in the Giffard injector, the pitting of boilers, the free access of water to propellers, the use of superheated steam, rotary engines, high pressure and other matters now held as settled, were by no means so held at the time of which we write.

For the information of very young engineers and the refreshing of some older ones, it may be recalled that doubts of surface condensation based their uncertainty upon the "crawling of materials," whereby condenser tubes moved until the ends were exposed, and not only so, but upon the "cussedness" of the material in not compensating the action by resuming the original position when cold.

As to the Giffard injector, now almost indispensable in locomotives and very useful for stationary boilers, the idea that a pipe leading from the top of a boiler to the water space of the same boiler, could feed water into it, seemed so repugnant to some Baltimore men (and they were supported by the bulk of European engineers) that they declared it impossible; and it should be recollected that the inventor, being unable to explain the action, contented himself by inviting unbelievers to come and see it in operation: then came the change—science having seen the fact, remembered that it was entirely according to physical law.

There are at least two mistaken ideas about the Engineer Corps of the Navy at the time herein referred to: one is that there was almost no information in print; the other that the older Chiefs, outside their profession, were almost ignorant men. As to the former, the status of marine engineering and the requirements of the service did not call for very much, but what was available was generally sufficient. We have mentioned the collection in manuscript of examination questions and answers, and certain books of the day were well

known and eagerly used; among which may be mentioned Byrne, followed by Bourne on the Steam Engine, Main and Brown on the Marine Steam Engine, Tomlinson's Mechanics, Gregory's Mathematics for Practical Men, Bonnycastle's (much-thumbed) Mensuration, Legendre's and other Geometries, Bourdon's and other Algebras, Eubank's Hydraulics, "as good as a novel," Lardner's somewhat discredited Natural Philosophy, Overman on the Manufacture of Iron, Johnson's Report on Coals, Nystrom's (the predecessor of Haswell's) Engineers' and Mechanics' Pocket Book, and several more; while in Baltimore at least, the libraries of some of the iron works, and more especially that of the Maryland Institute, contained books of great utility; the lectures of the latter on Chemistry and Philosophy were instructive, and its evening classes in drawing, etc., were open; engineers of the navy and of steam boats were always willing to impart the fruits of their experience; there were few mechanical periodicals, but those which existed brought new inventions and discoveries to notice and presented problems for study. Of course there was nothing like the mine of publications with which we are familiar now, with essays on every detail of every branch of marine engineering, and certainly such articles as the latest specifications in naval machinery and the papers found in the files of our JOURNAL, would have been invaluable to students in those early days, containing, as they do, the results of long labor in practice and theory.

The other statement, that the old chiefs could shove a file and chip to a line but knew little else, was also largely an error. Mr. Isherwood was well known to the whole scientific world, and others of his contemporaries were familiar with one or two languages or had dipped into the higher mathematics.

The Vulcan Works of Baltimore had done much naval work for the Government, including engines for the second *Princeton* and the *Susquehanna;* the firm competed during the time we are treating of, for the machinery of the *Dacotah,* one

of six vessels intended to be fast: the *Iroquois* was probably the best, but the Baltimore ship was not far behind her in speed and efficiency; however, the great weight of her geared engines and the care of Mr. Murray in making the details heavy enough to be safe, and perhaps the conscientiousness of the firm, contributed to the failure of the company in business. The *Dacotah* is worthy of notice for several reasons; among them the split connecting rod and main shaft revolving within it was a peculiarity, also the condenser had rather short perpendicular tubes, the upper ends expanded and the lower ones all packed with one follower-plate; some of the condenser features are almost obsolete now, but it was so successful in returning the water of condensation over and over again to the boilers that the latter developed one of the first mysterious cases of "pitting," an evil so little known at that time that it was universally attributed to the supposed presence of copper in the rain-water tanks at the Norfolk Navy Yard, where the vessel was fitted out. It was never dreamed that for years the scientific world, backed by committees appointed by the Lords Commissioners of the Admiralty, would later experiment upon and wrestle with the question of similar deterioration in boilers, and almost despair of its solution, in spite of at least three voluminous government reports in Great Britain, with colored plates, photographs and a wilderness of testimony, expert and otherwise.

Marine practise of the time referred to in our introductory remarks would have few advocates now, and deservedly so; the beam-engine was still holding its own, even for seagoing purposes; in boilers the advantages and disadvantages of copper and iron as materials, and of vertical and horizontal tubes, differential tubes, and even flues, were still discussed; Hall's, Sewell's, Stimers' and Pirsson's (double-vacuum) condensers were used, with others that are mere names; Stevens' and Sickel's cut-offs for poppet valves were best known; the Griffith, Hirsch, Mangin and other propellers were recommended, and some grades of naval engineers had to sketch and describe many devices which are forgotten now.

Drafting, previous to "the fifties" was ordinarily coarse in line, because among instrument makers even Hommel-Esser of Aargan in Switzerland, was only building up a reputation and his best productions (brass-mounted) were not pleasant to handle; but there were Department draftsmen in Washington, notably under the Lighthouse Board, who turned out beautiful work, which aroused admiration and emulation in the Baltimore contingent of the Engineer Corps, soon developing several young men skilled in the art. But the topic of drafting, in and out of the Navy, would itself need an article, and would testify to the excellence, among others, of George B. Whiting and John Ericsson, and their important relations in after years with the Bureau of Steam Engineering.

An account of Baltimore in some early years of the Engineer Corps, would not be complete without mention of the personnel who met in the drafting room of the Vulcan Works, many on the threshold of their career, but some even then far advanced in years and rank; two of the latter being known in warm admiration of their kind helpfulness as "Daddy" Williamson and "Pappy" Stewart, of Virginia and Pennsylvania, respectively. There was a spirit of comradeship among the aspirants, students and other hard workers for the few vacancies to be filled, but every one helping the whole by handing in his store of findings to the common stock. Mr. Bennett's list includes the whole Corps down to 1893, and he enlarges upon the Engineers-in-Chief and others of note, but without special points in reference to some of the juniors, and it is with apologies for necessary incompleteness, unintentional forgetfulness, etc., that we approach this part of the subject. Shock was already a well-known Baltimore name. Among the early Assistants best known, but even then by legend and anecdote, was J. Bolivar Knox, always called Bolivar, a name very fashionable at that time in honor of "The Liberator."

Ed. Chassaing was one of the brightest little men the Corps ever possessed, becoming fleet engineer in the civil war at an early age, and among other things pointing out an error in

King's "Steam," etc., upon the true reason why the salinometer stem was unequally divided; he resigned and later died of yellow fever in South America. Bob Schley and George Riley were authorities upon nautical affairs generally, having made voyages in sailing vessels; the former died in the service, the latter was afterwards connected with the Copper Works on the Gunpowder River in Maryland. Win. Thompson and a few others "seceded," and advanced in the Confederate engineer corps. Wilkins Cragg did not remain long in the service after the war, but Louis J. Allen did. Jim Noble was a sturdy, conscientious hard worker. Tom Bordley was a good draftsman and a good man. Tom Dukehart resigned. Phil. Voorhees was an accomplished engineer and gentleman, and later went into the patent law profession. John Wilson, genial and deservedly popular, resigned after the civil war. Al. Murray, a bright member of the Corps, was blown up in the *Chenango* boiler explosion, upon her trial. Nick Littig, light-hearted, seemed to make himself appear frivolous, but when, after the war, the *Oneida* was sunk in a collision, Chief Engineer Littig stuck to his post below and went down with the ship—a death as noble as man could wish.

Having mentioned briefly a few of the once well-known names of Baltimore naval engineers of the time specified, the writer feels it incumbent upon him to refer to a small number of a rather numerous class of civilian engineers, chiefs and assistants of seagoing and river steamers, superintendents and workmen in machine shops, and others from whose ranks the engineer corps obtained a goodly number of volunteers or acting appointments during the civil war, and who did excellent service in spite of the prejudices and teachings of their surroundings ashore and afloat: their history has perhaps never been, and probably never will be written, deserved though it is. Among a few names, but not all eligible for the Navy, may be mentioned Andy Miskimon, second to none when it was a question of running the lines in a hull and erecting engines therein; and here it may be interjected that there

was nothing in English print on the subject at that time. Other engineers were Tom Roberts, a willing helper; the older Gragg, a well-read man; Platt; Tim. O'Connor, who had run the first steam fire-engine in Baltimore; Rube McClanahan; the Lawrence and Thornton brothers; Bill Tipton, a finisher and afterwards an Acting Engineer in the Navy, who rallied a party of workmen from the Vulcan and Reeder Works, scaled Federal Hill just behind those shops, captured a cannon with which some fellows of the baser sort were firing a salute in honor of a Confederate performance, dumped the gun into the harbor, and showed that some Baltimore mechanics, at least, had not bowed the knee to the Baal of disunion; there were also John Cahill, John Menshaw, Bill Tolson and Tom Smith, the last the boss boiler maker.

Some events of the great war of 1914–18 may be recalled briefly and with a purpose, though not always having a bearing on naval engineering. The British have mentioned with deserved pride their saving of time by running troop trains directly on board special vessels which carried them across the Channel; and yet some Baltimore engineers will remember that before the fine bridges were built across the Susquehanna, the passenger trains to Wilmington were carried across the river upon large ferry boats. The use of parachutes has been spoken of as almost the last word in aviation, and yet, at the time covered by the title of this article some youths of Baltimore were thrilling over the fascinating book of John Wise, the Pennsylvania aeronaut, who, first in an emergency to avoid his balloon being carried out to sea, and afterwards by daring choice, turned his balloon itself into a parachute, by throwing out ballast until he was carried into such a rarified atmosphere that the balloon burst, and after a terrific fall of some distance, the bag, caught by the cords, became to some extent an umbrella and landed him safely; the flight across the Atlantic, now imminent and certain, was even then a theory.

We have read with pathetic interest the doings of faithful dogs and pigeons which carried war messages, and yet Scharf, in his Chronicles of Baltimore, relates how, first between that city and Washington, and later to Philadelphia and New York, a "pigeon express" was established before the telegraph superseded it.

Nowadays superheated steam is almost a commonplace necessity, and yet when the civil war broke out, an assistant engineer of the Navy was taking data and indicator cards by permission, on a boat running out of Baltimore, in order that the true economy in superheating should be tested. We boast now, and very justly, of carrying higher steam than our predecessors, and 400 pounds seem to be coming: at the time included in these notes, copper boilers still had their advocates, with ruder methods in riveting; but even then the book, Alban on the High-pressure Engine, was studied with an eye to the future, and there were 140-pounders on the Mississippi River. It is true that one of these would explode occasionally when an ambitious captain raced for the privilege of carrying on his steamboat the horns which denoted championship, but most human attainments must pay their price in human life. Torpedoes of a kind were known in our civil war, and registered a hit now and then. Oxy-acetylene welding almost worked miracles in repairing the machinery of interned men-of-war thought to have been irretrievably destroyed during the recent conflict, and yet it is many years since a United States naval engineer (from Baltimore, by the way) insisted that the pieces of a broken casting should be "burned" together instead of a new casting being made, though a well-known machine shop in the Far East declared that it had never heard of the process and would not be responsible for failure.

The turbine has almost revolutionized marine engineering, and yet rotary pumps were known in "the fifties" of the nineteenth century with yearnings toward less friction and waste of steam, while the pumps at least marked an improvement on the reaction globe of Hero of Alexandria. It must be con-

ceded, however, that the compound engine about to be revived was somewhat of a theory then, while multiple compounding, the electric drive, internal combustion, etc., were hardly in the mind of man. The comparative statements above are not intended to detract from modern achievements: the writer yields to none in profound admiration of progress in what is only the beginning of the twentieth century, but they are reminders of the old saw that "History repeats itself in the most unexpected circumstances," and they leave one in amazement of what the future may yet have in store for us.

The work of the writer is nearly finished: it is imperfect, rambling and inadequate, but if it will induce more competent hands to record the relations of various places to the early or late history of the Engineer Corps it will have accomplished good. And there is great store of interesting facts, from Baltimore, in extension of the foregoing lines; from Philadelphia, called slow but not so in engineering matters; from New York, based possibly on the local history by ex-Engineer-in-Chief Haswell, "the first engineer in the United States Navy," but to be rewritten with a slant toward the Corps; from Boston, for New England furnished us with some of our most intellectual members; from Washington, with wave upon wave of selected designers, draftsmen, calculators, workers and administrators; and from other localities, and the facts should be chronicled before it is too late.

But whatever may come, we are all confidently hopeful, as we have every reason to be, that the Engineer Corps of the U. S. Navy will live up to its traditions, and will continue to possess a Journal worthy to register its progress. May Corps and Journal be perpetual!

NOTES ON STEAM TRAPS.

By F. G. Hechler, Associate.

The requirements for general marine service are unusually severe; the trap must function satisfactorily at any pressure from atmospheric to the highest boiler pressure carried; its operation must not be adversely affected by the motion of the ship in a seaway; the *valve mechanism* must be simple and readily accessible for inspection and repair without having to disconnect any pipe line; the *discharge valve* must not be at the bottom of the trap where scale and sediment collect.

The body of the trap should be heavy and rigid, and flat surfaces should be avoided whenever possible. If the trap is intended for use with high steam pressures, the requirements of the American Standard for extra heavy flanged fittings should be complied with in determining the thickness of the body and of the cover flanges, as well as the size and the spacing of the cover bolts. Comparatively wide gasket spaces should be used to prevent the gasket from blowing out. Counter-bored recess gasket flanges are usually most satisfactory. The inlet and outlet opening should be on the body of the trap so the cover may be removed without breaking any pipe lines, and there should be a bottom blow-off for cleaning the trap of scale and sediment.

In ball float traps the buoyancy of the float must be sufficient to open the valve against the steam pressure. In this case the following relation must be satisfied:

$$W + (p_s - P_d)\, a < F,$$

Where W = weight of float in pounds p_s = steam pressure in trap, pounds per square inch gage; p_d = pressure against which trap discharges (usually atmospheric), pounds per square inch gage; a = contact area of discharge valve, square

inch, F = effective buoyancy of float = displacement in cubic feet times density of water in trap at temperature corresponding to the pressure, pounds per cubic feet.

A 1¼-inch trap of the simple float type and intended for use with pressures up to 300 pounds, had a float displacement of 0.1278 cubic foot; the float weighed 4.17 pounds. The maximum size of the discharge valve calculated from the relation just given is $4.17 + (300 - 0)\ a < 0.1278$ times 52.6, or $a < 0.0085$ square inch, and the corresponding valve diameter is 0.104 inch.

Small discharge valves are unsatisfactory because they are liable to become clogged with scale and because their discharge capacity is too small. Large floats are not desirable because they increase the size of the trap unduly, and because it is difficult to make a large float to withstand the high pressures. For these reasons the use of levers to increase the float pull is usually resorted to.

The effective pull of a bucket float for opening the valve may be found by filling the trap and bucket with water, attaching a spring balance to the valve stem and measuring the pull necessary to lift the bucket. Tests have shown that for bucket floats the relation between maximum operating pressure, valve area and bucket pull may be written as $p_s \times a^n =$ F, where p_s a and F have the meanings already given, and where the exponent n has a value that is usually slightly less than unity (results varying from 0.93 to 1.00 have been found). When graphed on logarithmic paper this curve is a straight line whose slope gives the value of the exponent n. The relations between p_s and a as determined experimentally for a line of bucket-float traps varying in nominal size from ¾-inch to 2-inch are shown in Fig. 1. The p intercept for an area of 1 square inch is the pulling power of the bucket as determined with the spring balance. In practice the valve areas should be somewhat less to insure satisfactory operation for slight over-pressures that may occur, and also to per-

mit grinding in the valve and trueing up the seat when necessary. The allowance in area should be from 10 to 15 per cent; valve areas determined in this manner will be consistent

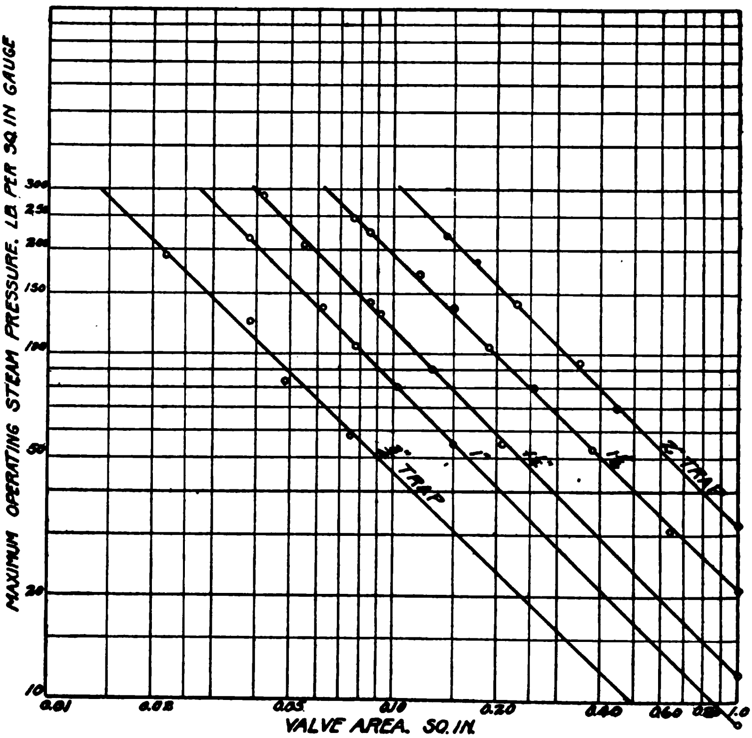


FIG. I.

and will give the maximum safe capacity. The design of the bucket float requires care; the addition of a small weight at the outer end of the bucket may increase the maximum operating pressure of the trap with a given valve as much as 20 per cent.

TRAP RATINGS AND CAPACITIES.

There is no logical reason why steam traps should be rated, and also usually purchased, on a basis of the size of the inlet and outlet connections. Traps of the same nominal size may

have widely different capacities at the same pressures as shown in Table 1. When specifying capacity it is necessary to state the pressure at which it is required. Capacities should always be determined with hot water delivered to the trap by steam at the necessary pressure; capacities determined with cold water and a pump will be too high.

Steam traps (except expansion) will operate satisfactorily at any pressure lower than the designed pressure without change or adjustment, but with a smaller capacity. To secure maximum capacity the valve should always be adapted to the operating pressure. When a trap is likely to be used over a considerable range of pressures it should be provided with several interchangeable valves of different size. For pressures up to 300 pounds it has been found sufficient to use three valves and seats designed for pressures of 30, 150 and 300 pounds. Table 2 gives the capacities as determined by test of a line of bucket traps at pressures of 30, 100 and 200 pounds. These capacities are the maximum quantity the traps can pass when discharging continuously under full steam pressure.

INSTALLATION, CARE AND OPERATION.

When installed on shipboard, traps are seldom horizontal, and the constant change in inclination, due to the pitching and rolling of the vessel in a seaway, may interfere with their action. Many of the bucket and ball-float traps lose their water seal and open the discharge valve, permitting steam to blow through when inclined in a longitudinal direction through an angle of about 15 degrees with the horizontal. This difficulty is to some extent obviated by making the operating mechanism as compact as possible; it has also been found with bucket traps that it is possible to increase the safe angle of inclination up to about 25 degrees by making the bucket with an inclined upper edge similar to that shown in Fig. 4, page 244, JOURNAL A. S. N. E., May, 1918. Whenever possible,

traps should be installed on shipboard with their longitudinal axis fore and aft, thus decreasing the liability of steam blowing through. The operation of the traps shown in Figs. 6 and 7, page 243, JOURNAL, A. S. N. E., May, 1918, is not affected by any inclination likely to occur. Expansion traps are also free from this defect as they operate equally well when installed in any position.

When the pressures are subject to considerable variation, expansion traps are not recommended for use unless provided with an automatic pressure-compensating device. With some traps of this type it is difficult to maintain a given setting even under constant pressures, and in some cases the temperature of the surrounding air has an appreciable effect on the operation. Expansion traps may be adjusted to give either a constant or an intermittent discharge; they do not become air-bound, but sediment and grease prove troublesome. Since they have no storage capacity, 8 or 10 feet of pipe should be provided for cooling the condensate somewhat before it reaches the trap.

When continuous-discharge traps are operating at low capacities the *wire drawing* at the valve is likely to prove destructive to the valves and seats; these should be arranged so they may be readily examined and renewed when necessary. The continuous-discharge trap is well suited for use with long discharge lines, preventing water hammer and sudden shock likely to occur with the intermittent type under this condition. Whenever possible traps should be located so that the condensate will flow to them by gravity. If such a location would make the trap inaccessible, it may be placed at a higher level if the line to be drained carries a pressure of 5 pounds or more. The drain line should always be connected to the lowest part of the line or apparatus to be dripped. A trap should be connected to drain only a single line unless the pressures are known to be identical; this is a rare condition and is likely to be disturbed from time to time. Traps may

discharge to a higher level than the trap if necessary; it is usual to assume that the condensate may be elevated two feet for each pound of excess pressure in the trap over that in the discharge line. Traps should always be connected up with unions and fitted with a by-pass so that the trap may be overhauled without interfering with the proper drainage of the line. A test valve or cock should be placed in the discharge line close to the trap to determine how the trap is functioning. A blow-off line not smaller than ½ inch should be provided for cleaning the trap.

A regular *inspection* of all traps should be insisted upon, and they should be promptly overhauled when improper functioning is detected. Steam traps are prone to leak due to cutting of the seat or scale lodging under the valve. Because under these circumstances there is no difficulty due to an accumulation of water, repairs are often neglected, permitting a serious steam leakage to occur.

FITTINGS.

Gage glasses are desirable aids for determining the condition of a steam trap. Under high steam pressures tubular gage glasses sometimes break frequently so that they are often dispensed with entirely. Gage glass fittings should be of the ball-check type so they will be self-closing in case of accident. An air cock is necessary with most of the float traps to relieve the air when starting, and at such other times as may be necessary. A removable strainer placed at the trap inlet, or in the pipe line just before the trap, is often of value in preventing scale and sediment reaching the working parts of the trap.

TESTING.

No uniform code for *testing* steam traps has been adopted. New types of steam traps submitted for use in the Naval Service are subjected to the following tests (see JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS, Vol. 28, No. 1,

February, 1916): (a) *Hydrostatic pressure* test of 150 per cent of the maximum working pressure (ball floats are separately tested to 500 pounds per square inch. (b) *Functioning* test with steam pressures from 25 to 300 pounds gage. (c) *Capacity* test to determine the maximum quantity of water the trap can discharge under various steam pressures. (d) *Rolling* test with the trap mounted on a rocking platform to simulate the motion of a vessel in a heavy seaway. (e) *Endurance* test with the trap installed for at least three months in the power plant to determine whether it will function properly for a reasonable length of time.

The specifications of the Bureau of Steam Engineering, Navy Department, for steam traps are as follows: (1) Traps will be of either open-bucket or ball-float positive action type, unless otherwise specially approved by the Bureau, so designed that they can be overhauled without disconnecting the pipes, and the bonnets removed without disturbing the lagging. They will be located where they are easily accessible for examination; (2) the trap must be of a type that has given satisfaction in actual service on board a Naval vessel, or passed a satisfactory test at the U. S. Naval Engineering Experiment Station, Annapolis, Md. If a ball float is used the ball must be able to stand an external pressure of not less than 500 pounds per square inch; (3) the blow-off will be arranged for hand operation, and must have a free opening for discharge, not less than ½ inch in diameter. The drain (discharge valve) must be high enough above the bottom of the trap to prevent being clogged with sediment. All pipe connections to traps will be made with union couplings or by flanges. All traps will have by-pass pipes and valves for convenience of overhauling. All traps will be fitted with valves for testing.

TABLE 1.

Discharge Capacity of Steam Traps.

Make of Trap.	Type.	Nominal size.	W'ght lbs.	Capacity at 250 lbs. pressure.	Remarks.
Vance	Bucket	1-¼″	145	26,380	Compound valve.
Ideal	Ball float	1-¼″	280	7,040	
Kieley & Mueller	Bucket	1-½″	200	20,250	Compound valve.
Sarco	Expansion	¾″	14	2,700	At 200 lb. pressure.
Johns-Manville	Ball float	1-¼″	44	3,480	
Armstrong	Inverted bucket	1″	67	3,200	
Anderson	Ball float	1-¼″	165	6,230	
Corliss Valve	Ball float	1-¼″		4,250	
Swartwout	Bucket	1-¼″	145	4,100	
McDaniel	Ball float	1-½″	136	13,800	
Strong	Bucket	1-¼″	176	4,800	
Fisher	Bucket	1-¼″	151	4,800	
Wright	Ball float	1-¼″	109	7,350	Three valves opening consecutively.
Sterling	Ball float	1″	84	3,600	

TABLE 2.

Variation of Capacity with Size of Trap.

Nominal size, Diameter inlet and outlet, inches.	Capacity, pounds per hour under steam pressures of; pound per square inch gage.		
	30	100	200
¾	6,100	4,400	2,400
1	8,600	8,200	6,400
1¼	14,600	9,900	9,100
1½	12,600	13,700	15,500
2	17,600	25,400	25,400

The above capacities were obtained by using an appropriate valve for each steam pressure.

TEST OF MAIN CIRCULATING PUMP OF U. S. S. *NEW MEXICO.*

BY J. S. EVANS, COMMANDER, U. S. N., MEMBER.

On account of limited facilities at the works of the manufacturer it was not practicable to test the main circulating pumps of the *New Mexico* before installation in the ship and, as the contract specified pumps of a certain capacity against a certain head, it was necessary to test the pumps after installed. As these pumps are motor driven it was possible to measure the power input to the pumping set much more accurately than on any previous test on board ship, this on account of the much greater accuracy of electrical measurements over those obtained from the steam-driven units which had formerly been installed on shipboard. From the tests made on the motors during inspection at the works of the motor manufacturer the efficiency of the motor at various speeds was known and the power input to the pump was thus easily obtained.

The static head was measured at six points as shown in Figure 1, the location of points being as follows:

Point No.	*Location.*	*Height above base line.*
1.	In main injection pipe........	5 feet 6 inches.
2.	In pump discharge near pumps..	12 feet 3 inches.
3.	At entrance to main condenser..	20 feet 10½ inches.
4.	At middle of rear water chest of condenser	22 feet 6 inches.
5.	At outlet of main condenser....	22 feet 2 inches.
6.	In discharge pipe from condenser	28 feet 5 inches.

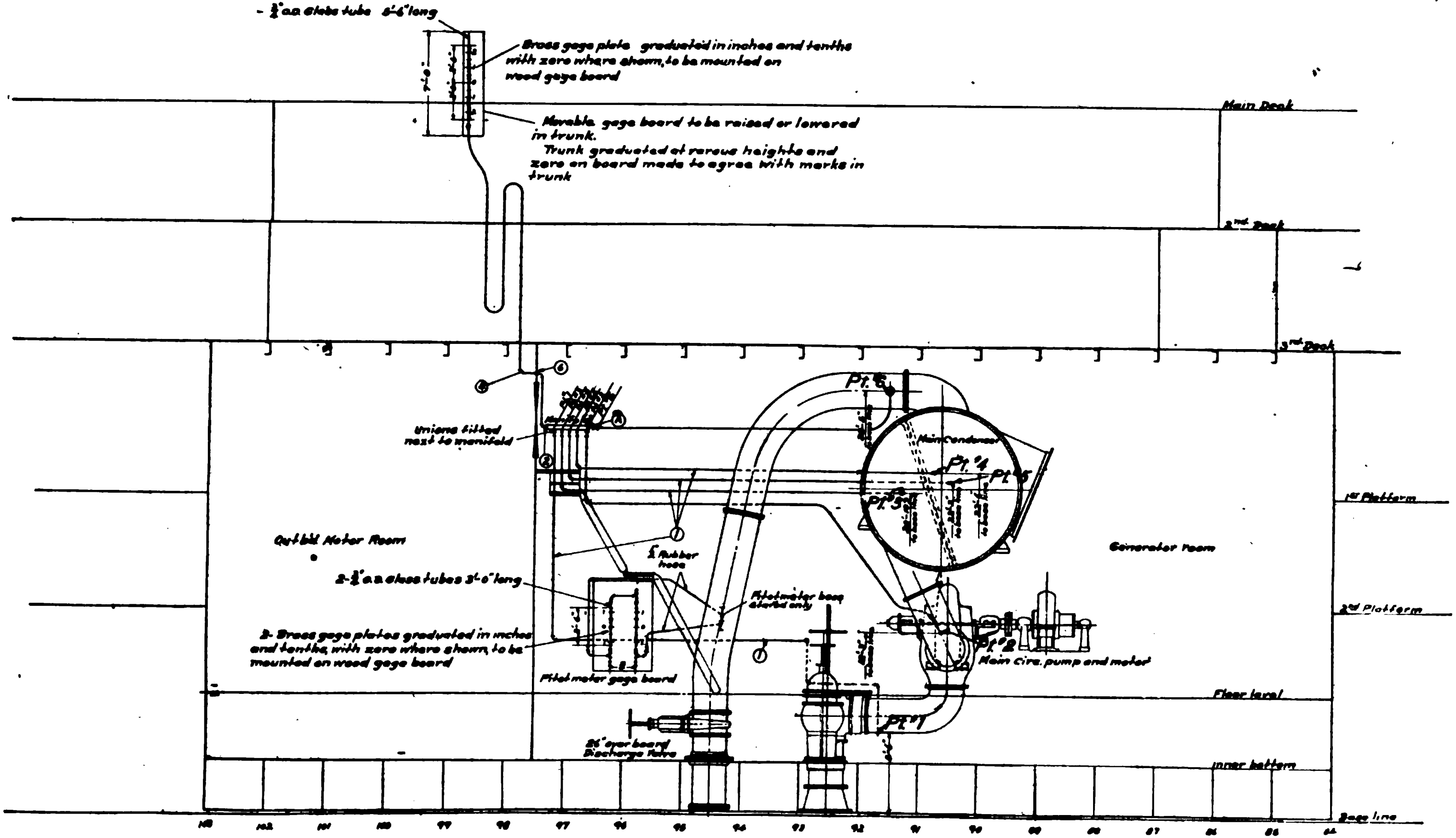
The water line was 30 feet 9 inches above base line. Figure 1 shows the piping from the above-mentioned points. One-half-inch pipes were led from each of the points to a manifold just under the third deck, each line being provided with a valve at the manifold. From this manifold a line of ½-inch

pipe and flexible hose was led to a gage board which was so arranged that it could be raised or lowered in a motor-room air trunk, and this trunk was graduated at various heights. For taking readings the zero of the gage board was made to coincide with a graduation on the trunk, and it was therefore possible to obtain the exact height above the base line of water level in the glass. All readings obtained are given in Table No. 1, columns 8–13, and are measured from the base line. The reading at each point for each speed was obtained by opening the valve at the manifold in the line leading from that point, with valves in the other lines closed, and all readings were checked until as accurate as possible. However, no readings at any speed were taken until the speed of the motor was constant at the speed set, and there was very little fluctuation in the gage glass.

In making calculations the readings from point No. 2 were disregarded, as it was believed from the beginning of the test that the readings from this point would be in error, this on account of the fact that the static-pressure tube used here had to be put in at a point where the piping between pump and condenser made a 90 degrees bend, the pressure tube being inserted in this piping through a boss which ordinarily is used for a fixed thermometer.

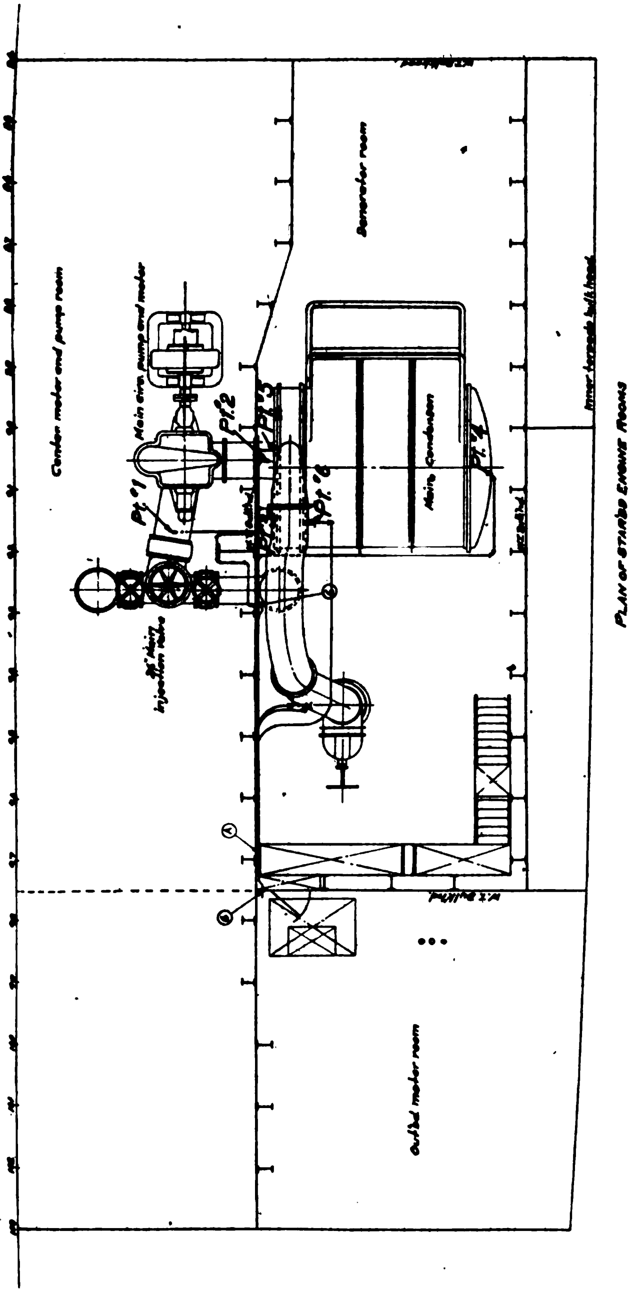
The velocity head was measured in the overboard discharge pipe by means of two velocity head tubes, approximately 90 degrees apart, two tubes being necessary on account of the fact that there was at no place in the piping a straight section of sufficient length to insure an even flow. The arrangement of the gage board and of the piping and hose from the tubes is shown on Figure 1, an air cock being provided at the top of the gage glasses to permit of drawing off air so that the readings would come on the gage board. Graduated brass scales were secured directly in rear of each gage glass.

The tubes used consisted of an impact tube surrounded on its straight portion by a static tube, the connections from each being brought out separately, the connection from the static



ELEVATION AT STARBD. SIDE LOOKING INBOARD

FIG. 1.



PLAN OF STARBD ENGINE ROOMS

FIG. 1.

tubé being led to one leg of the gage and that from the impact tube to the other leg. Valves were fitted in each connection.

The ten-point method was used in getting the velocity head readings; thus, with the two velocity head tubes twenty readings were obtained for each speed of the pump and the square of the average square root of these readings was taken as the velocity head at that speed.

Before starting the test the pump was run under load until the motor was at a constant temperature, this being not only the usual running condition, under which condition it was desired that the pump be tested, but also the condition under which the efficiency tests of the motor were made.

The data obtained on the test is shown in Table No. 1, columns 1, 2, 3, 6, and 8 to 13, inclusive, but the data in column 9 was disregarded for reasons previously explained. Columns 4, 5, 7 and 14 give the data calculated from that in the other columns, the capacities in column 7 being calculated from the formula: Capacity in gallons per minute $= \dfrac{8\sqrt{12h} \times 530.9 \times 60}{231}$, where h equals velocity head in inches of water, and 530.9 is the cross sectional area of the 26-inch overboard-discharge pipe. The total heads as given in column 14 were obtained by taking the difference in readings between points 1 and 3, there being no loss between these two points except the small amount of loss of head due to friction, which is disregarded in the calculations, and the velocity heads at these points cancel each other since the injection and discharge pipes are of the same diameter. The data given in columns 4, 5, 7 and 14 are plotted on Figure 2, fair curves being plotted. From these curves the data given in Table No. 2, columns 1, 2, 3 and 5 were picked off for speeds varying by 50 r.p.m., but it must be understood that any data or results for speeds lower than 200 r.p.m. are only approximate, that being the lowest speed used on the test. The efficiency of the motor as given in column 7 of Table No. 2 was obtained from acceptance test runs made on the motor at the works of

TABLE No. 1

1	2	3	4	5	6	7	8	9	10	11	12	13	14
R.P.M.	VOLTS	AMPS.	K.W.	H.P.	h Velocity head in inches of water	Q Capacity in G.P.M. [illegible]	Reading of water column in feet at point #1	#2	#3	#4	#5	#6	H Total head in feet of water Col. 10 – Col. 8
200	118	135	15.9	21.4	5.14	8670	24.21	28.25	29.63	27.71	26.46	25.33	5.42
245	118	200	23.6	31.6	7.20	10260	23.71	31.04	31.50	28.63	26.96	25.58	7.78
290	118.	320	37.8	50.6	10.22	12220	23.21	32.63	33.63	29.92	27.67	25.58	10.42
335	115.5	505	58.3	78.2	13.95	14275	22.04	35.29	36.54	31.46	28.38	25.63	14.50
405	237	440	104.3	139.8	21.28	17630	21.00	39.96	41.96	34.38	29.83	25.79	20.96
440	237	505	119.7	160.4	24.16	18785	20.38	41.46	43.88	35.54	30.63	25.83	23.50
500	237	760	180.1	241.5	31.63	21500	19.21	46.13	52.80	38.63	32.25	26.21	33.59
518	237	880	208.6	279.6	34.72	22520	17.79	53.64	54.01	39.79	32.38	26.50	36.22

TABLE No. 2

1	2	3	4	5	6	7	8
R.P.M.	Q Cap. in G.P.M.	H Total head in feet	W.H.P. $\frac{Q \times H}{3895}$	H.P. input to set	Efficiency of Set	Motor	Pump
50	2140	0.38	0.2	2.5	.08	.19	.42
100	4275	1.3	1.42	7.0	.20	.375	.54
150	6425	2.8	4.62	13.8	.342	.565	.605
200	8575	5.0	11.01	22.25	.495	.755	.655
250	10700	8.0	21.98	36.00	.611	.858	.712
300	12850	11.6	38.27	58.00	.66	.885	.746
350	15000	15.8	60.85	88.00	.691	.89	.777
400	17150	20.8	91.58	128.00	.715	.888	.806
450	19275	26.4	130.64	178.00	.734	.91	.806
500	21450	33.5	184.48	245.00	.753	.9184	.82
518	22200	36.22	206.44	279.00	.739	.9185	.804

TABLE No. 3

1	2	3	4	5	6	7	8	9	10	11
R.P.M.	Reading at point #1	#3	#4	#5	#6	26.25 feet minus reading at point No. 1	Difference between readings at points #3 & #4	#4 & #5	#5 & #6	Reading at point No. 6 minus 25.25 feet
200	24.21	29.63	27.71	26.46	26.33	1.04	1.92	1.25	1.13	0.08
245	23.71	31.50	28.63	26.96	25.58	1.54	2.87	1.67	1.38	0.33
290	23.21	32.63	29.92	27.67	25.58	2.04	3.71	2.25	2.09	0.33
335	22.04	36.54	31.46	28.38	25.63	3.21	5.08	3.08	2.75	0.38
405	21.00	41.96	34.38	29.83	25.79	4.25	7.58	4.55	4.04	0.54
440	20.38	43.88	35.54	30.63	25.83	4.87	8.34	4.91	4.80	0.58
500	19.21	52.80	38.63	32.25	26.21	6.04	14.17	6.38	6.04	0.96
518	17.79	54.01	39.79	32.38	26.50	7.46	14.22	6.91	6.38	1.25

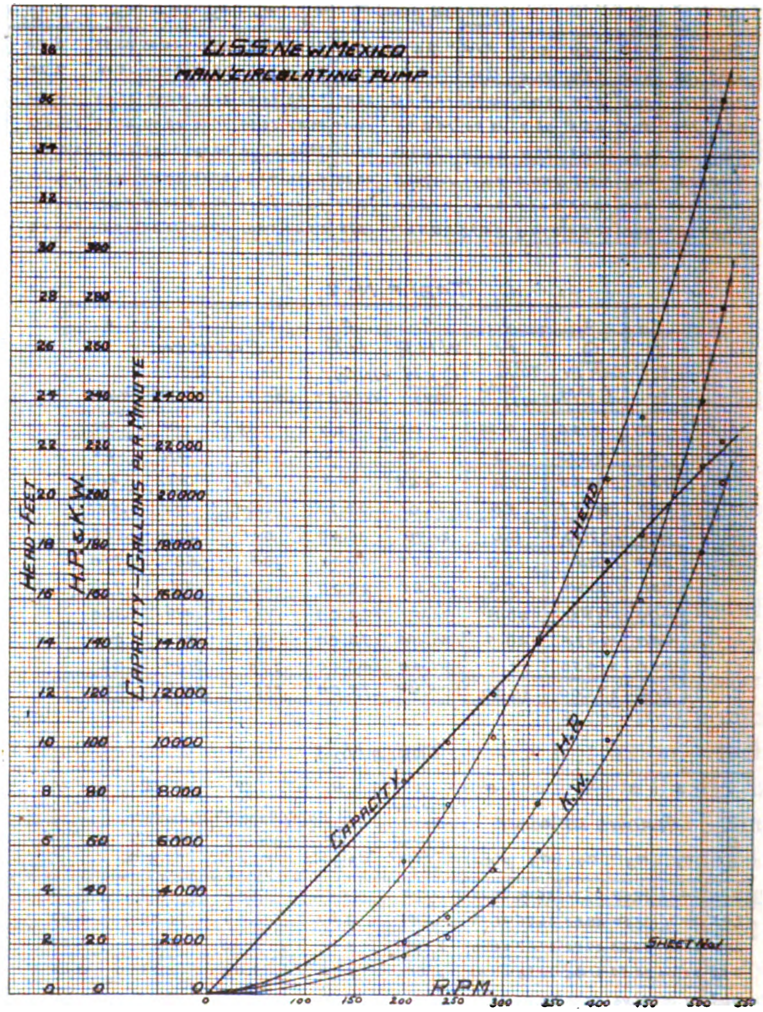

U.S.S. New Mexico
Main Circulating Pump
Head-Feet
H.P. & K.W.
Capacity-Gallons per Minute
Capacity
Head
H.P.
K.W.
R.P.M.
Sheet No. 1

FIG. 2.

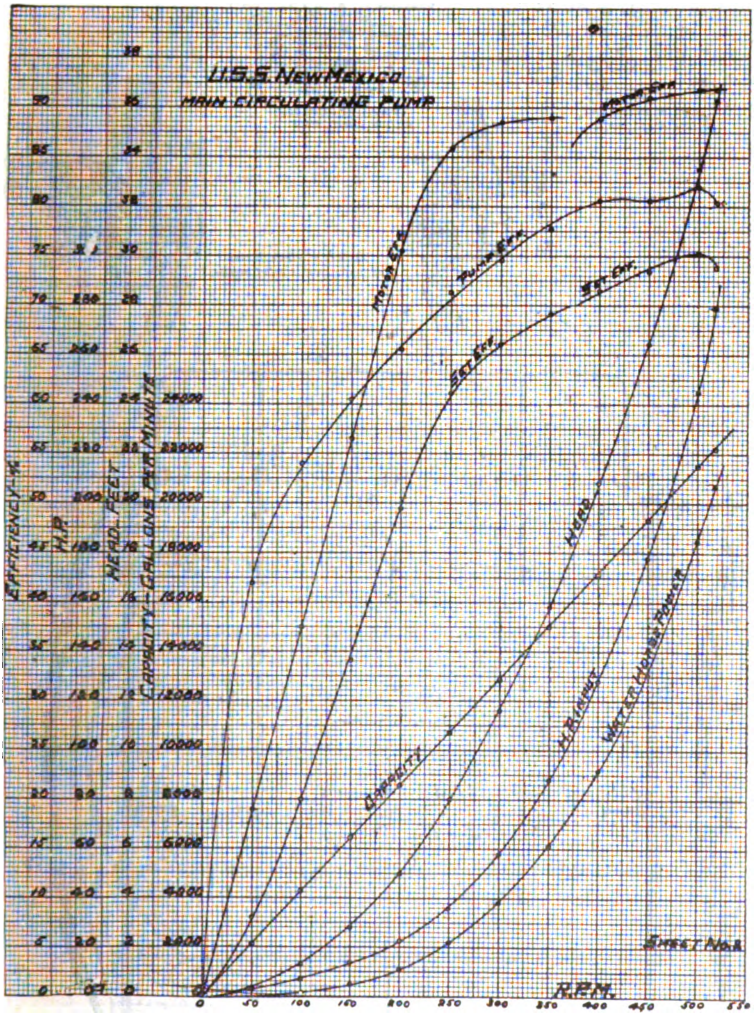

U.S.S. New Mexico
MAIN CIRCULATING PUMP
Efficiency-%
H.P.
Head, Feet
Capacity-Gallons per Minute
Motor Eff.
Pump Eff.
Set Eff.
Head
Capacity
H.P. Input
Water Horse Power
Sheet No. 2
R.P.M.

FIG. 3.

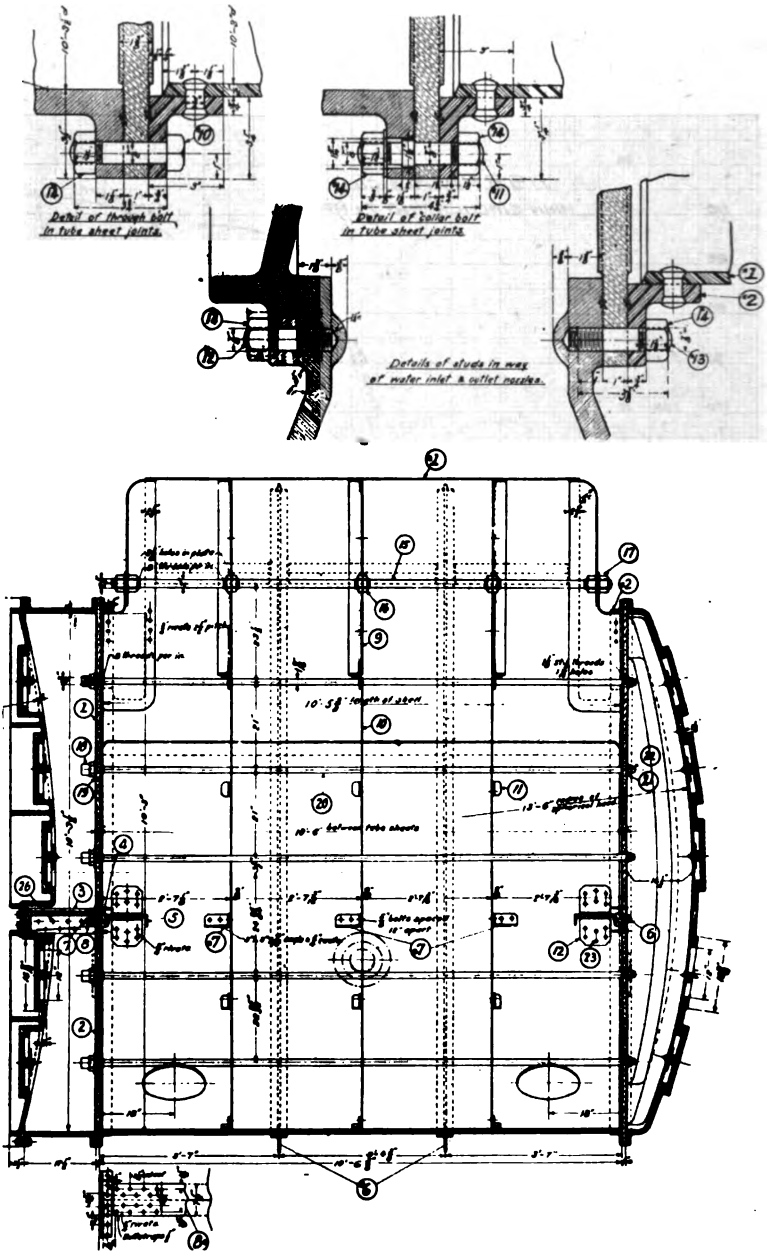

Detail of through bolt in tube sheet joints.
Detail of collar bolt in tube sheet joints.
Details of studs in way of water inlet & outlet nozzles.
10'-5⅝" length of shell
10'-6" between tube sheets

FIG. 4.

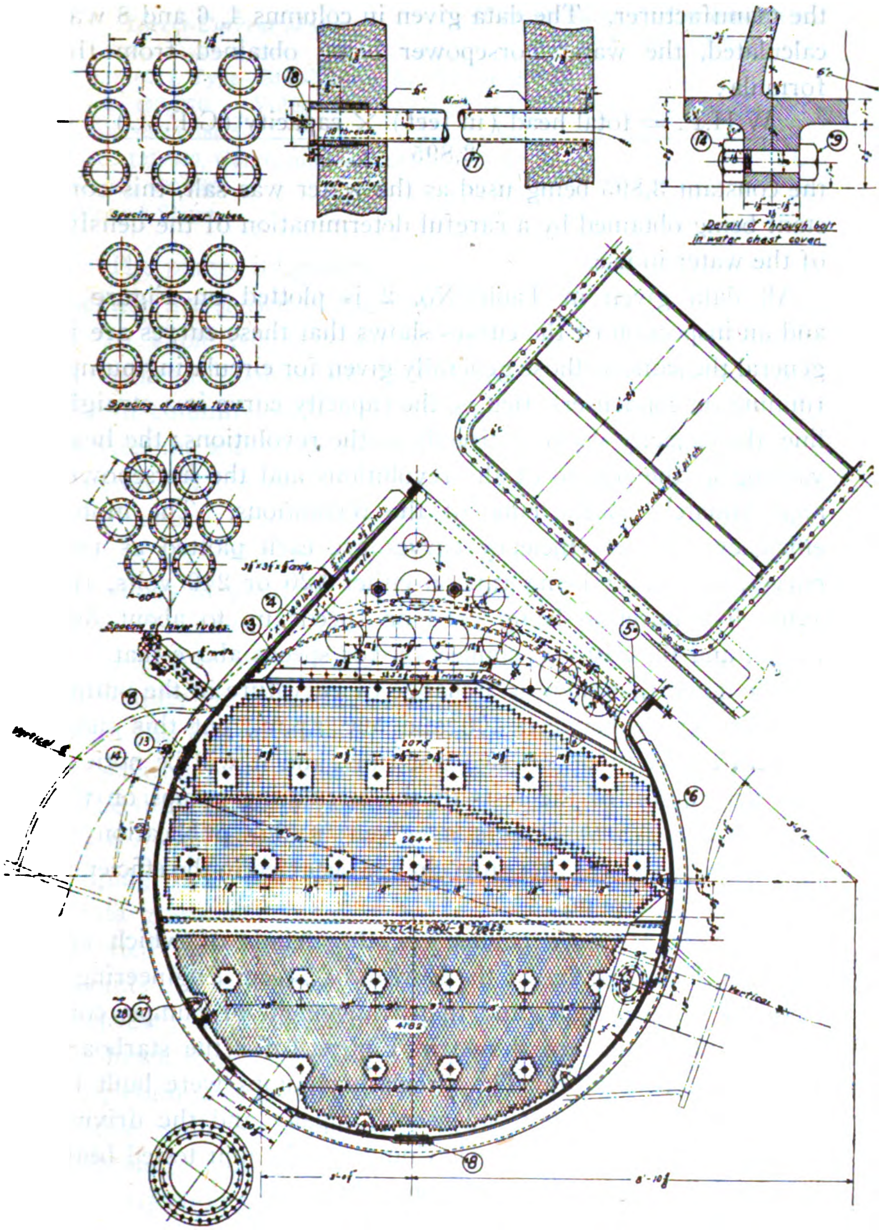

Spacing of upper tubes.
Spacing of middle tubes.
Spacing of lower tubes.
Detail of through bolt in water chest cover.
Water outlet side.
2075
2644
4182

FIG. 4.

the manufacturer. The data given in columns 4, 6 and 8 was calculated, the water horsepower being obtained from the formula:

$$\text{W.H.P.} = \frac{\text{total head (in feet)} \times \text{capacity (G.P.M.)}}{3{,}895}$$

the constant 3,895 being used as the water was salt, this constant being obtained by a careful determination of the density of the water in use.

All data given in Table No. 2 is plotted on Figure 3 and an inspection of the curves shows that these curves are in general the same as those generally given for circulating pumps running on condensers, that is, the capacity curve is a straight line, the capacity varying directly as the revolutions; the head varying as the square of the revolutions and the horsepower approximately as the cube of the revolutions. The motor efficiency and set efficiency curves are each plotted as two curves, the motor being run on either 120 or 240 volts, the 120-volt connection being used on speeds up to about 360 r.p.m., and the 240-volt connection for speeds above that. It will be noted that there is a change of curvature in the pump-efficiency curve between 400 and 500 r.p.m., but this same curvature was found on two test runs, and it will be noticed in a description of the tests of the circulating pumps on the *Nevada* and *North Dakota* as given in this publication in February, 1918, that this same change occurred in the efficiency curves of these pumps.

The *New Mexico's* condensers, the details of which are shown in Figure 4, are of the Bureau of Steam Engineering's standard circular type, and the arrangement of pumps, condenser and piping are shown in Figure 1 for the starboard main turbo-generator. The circulating pumps were built by the Alberger Pump & Condenser Company and the driving motors by the General Electric Company, the unit tested bearing the following data on the name plates:

Direct-Current Motor.	*Alberger Pump & Condenser Co., 24-inch Volute Pump.*		
No. 616244, Type—MPC, Form A.	No. 8692.		
Amp. 450/870 Volts, 230.	*Cap. G.P.M.*	*Speed, R.P.M.*	*Head, Feet.*
K.W. 93.25/186.5.	10,000	270	9.5
H.P. 125/250, Continuous service.	16,000	415	21.0
Speed, 400/500.	19,000	500	30.0
Wound Shunt.	21,600	560	36.0

The pump, its general dimensions, and elevation and cross section are shown in Figures 5 and 6.

In Table No. 3 is an analysis of the various parts of the total head. Columns 2, 3, 4, 5 and 6 of this table are copied from columns 8, 10, 11, 12 and 13 of Table No. 1. The water line was 25.25 feet above point No. 1, so the differences between 25.25 feet and the readings at this point represent the suction head on the pump at the various speeds; these heads are given in column 7. The differences between the readings at points 3 and 4 and at points 4 and 5 show the loss of head in the lower half and upper half of the condenser respectively, and are tabulated in columns 8 and 9, respectively. The differences between the readings at points 5 and 6 represent the loss in the exit nozzle and a 90-degree bend in the discharge pipe, these differences being given in column 10. The differences between 25.25 feet and the readings at point No. 6 represent the loss of head in the overboard-discharge piping, these differences being tabulated in column 11. The data given in columns 7 to 11, inclusive, Table No. 3, was plotted on Figure 7, and curves were drawn through the points. It will be noticed that of curves Nos. 1 and 6, which represent heads due principally to resistance from bends, No. 6 is much lower than No. 1; No. 1 being the resistance of the injection piping and valve which is a 26-inch angle-globe valve, while No. 6 represents the resistance of the bends of the discharge piping, which bends are much easier than those in the injection piping, the resistance of discharge valve, which is a 26-inch gate valve as against an angle-globe valve in the injection line, and the resistance of the water leaving the ship's

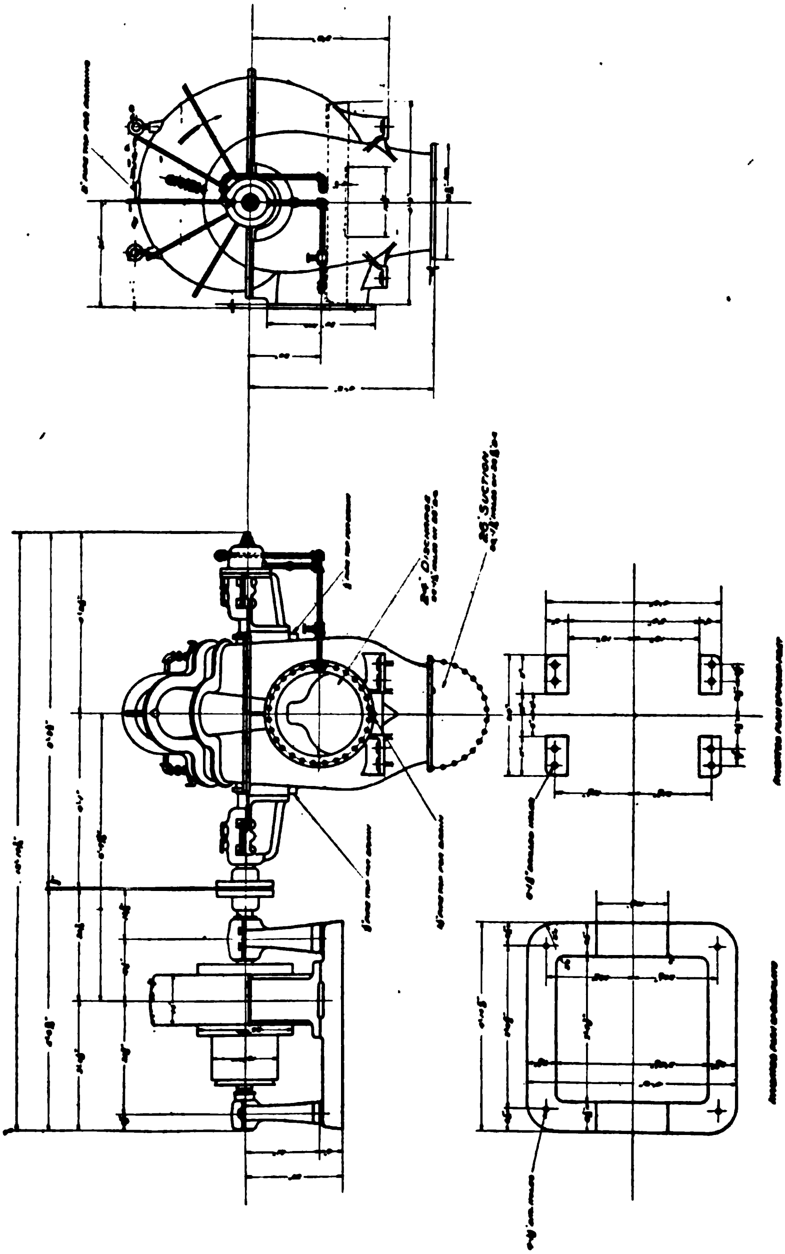


FIG. 5.

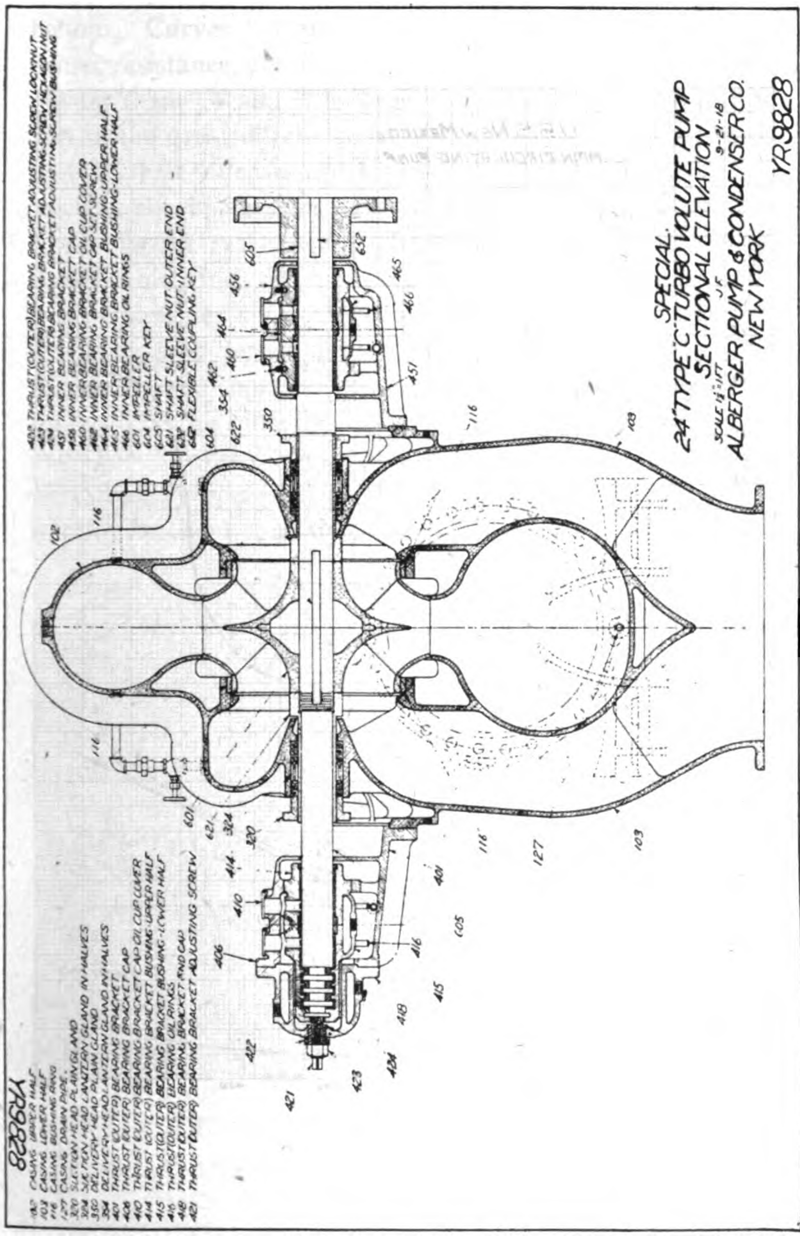

YR9828
102 CASING UPPER HALF
103 CASING LOWER HALF
116 CASING BUSHING RING
127 CASING DRAIN PIPE
320 SUCTION HEAD PLAIN GLAND
324 SUCTION HEAD LANTERN GLAND IN HALVES
350 DELIVERY HEAD PLAIN GLAND
354 DELIVERY HEAD LANTERN GLAND IN HALVES
401 THRUST (OUTER) BEARING BRACKET
406 THRUST (OUTER) BEARING BRACKET CAP
410 THRUST (OUTER) BEARING BRACKET CAP OIL CUP COVER
414 THRUST (OUTER) BEARING BRACKET BUSHING-UPPER HALF
415 THRUST (OUTER) BEARING BRACKET BUSHING-LOWER HALF
416 THRUST (OUTER) BEARING OIL RINGS
418 THRUST (OUTER) BEARING BRACKET END CAP
421 THRUST (OUTER) BEARING BRACKET ADJUSTING SCREW
422 THRUST (OUTER) BEARING BRACKET ADJUSTING SCREW LOCKNUT
423 THRUST (OUTER) BEARING BRACKET ADJUSTING SCREW HEXAGON NUT
424 THRUST (OUTER) BEARING BRACKET ADJUSTING SCREW BUSHING
451 INNER BEARING BRACKET
456 INNER BEARING BRACKET CAP
460 INNER BEARING BRACKET OIL CUP COVER
462 INNER BEARING BRACKET CAP SET SCREW
464 INNER BEARING BRACKET BUSHING-UPPER HALF
465 INNER BEARING BRACKET BUSHING-LOWER HALF
466 INNER BEARING OIL RINGS
601 IMPELLER
604 IMPELLER KEY
605 SHAFT
621 SHAFT SLEEVE NUT OUTER END
622 SHAFT SLEEVE NUT-INNER END
652 FLEXIBLE COUPLING KEY
SPECIAL.
24" TYPE "C" TURBO VOLUTE PUMP
SECTIONAL ELEVATION
SCALE 1½"=1FT J.F. 9-21-18
ALBERGER PUMP & CONDENSER CO.
NEW YORK
YR9828

FIG. 6.

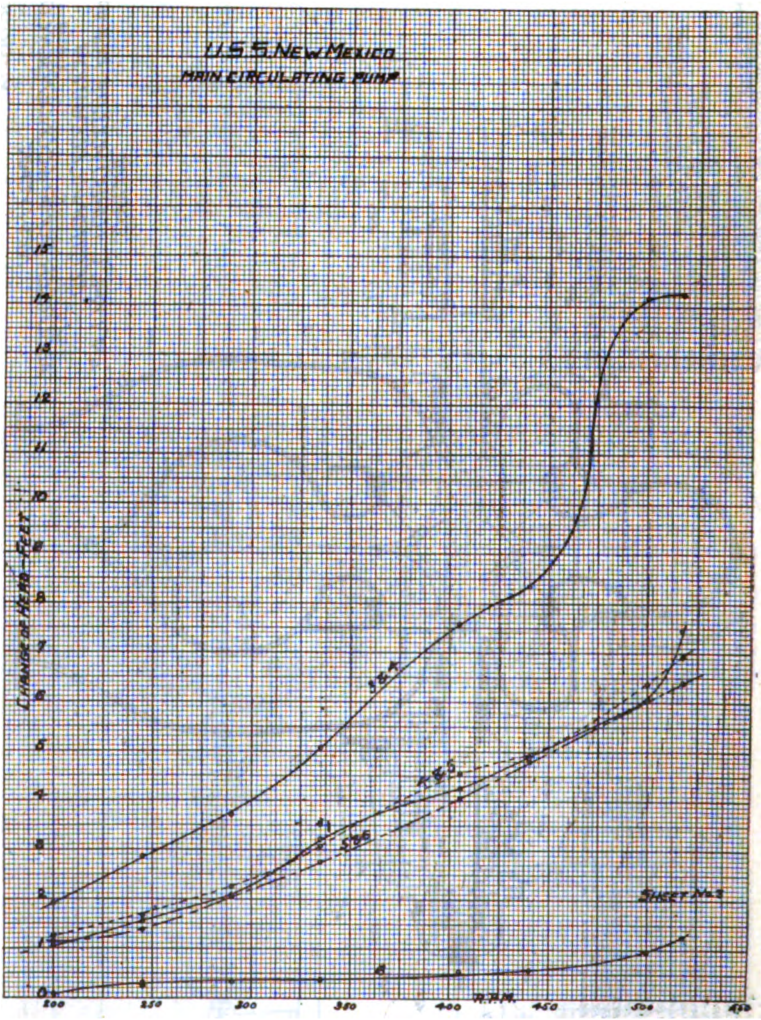

U.S.S. New Mexico
Main Circulating Pump
Change of Head—Feet
R.P.M.
Sheet No. 3
200
250
300
350
400
450
500

FIG. 7.

bottom. Curves "3 and 4" and "4 and 5" represent condenser resistance, and it will be noted that "3 and 4" is much greater than "4 and 5." This is undoubtedly due in a large part to the great difference in the number of tubes in the two parts of the condenser, the lower part having only 4,182 tubes, whereas the upper part has 4,719 tubes. Curve "5 and 6" represents the resistance of the discharge nozzle of the condenser which includes a short-radius 90-degree bend.

Several times since the above test was run the speeds of the pump and the ampères input have been checked against the speeds and input as used on test, and the test conditions found to be the same as the usual running conditions. These units have operated very satisfactorily, and when both pumps were recently opened for examination it was found that they were in excellent condition.

THE WILLIAMS REFRIGERATING MACHINE.

WM. L. DEBAUFRE.

A test of the Williams refrigerating machine, manufactured by the Electrical Refrigerating Co., Inc., New York City, was made at the Naval Engineering Experiment Station, Annapolis, Md., under authorization of the Bureau of Steam Engineering of the Navy Department. A description of the machine with some of the test results follows:

DESCRIPTION OF MACHINE.

A sectional elevation of the machine appears in Fig. 1 and a sectional plan in Fig. 2. A general view of the machine is shown in Fig. 4 and the compressor parts are shown in Fig. 5.

The operating parts of the machine, including the rotary vapor compressor, the vapor condenser, the refrigerant-control valve, etc., are all inclosed within a vertical cylindrical casing having a dome-shaped head. This casing serves as a reservoir for the refrigerant (ethylchloride) and the lubricant (glycerin), and also forms a jacket around the compressor for the cooling water. The shaft of the compressor projects through a stuffing box in the casing and is driven by an electric motor mounted on a common base plate. Automatic-control devices are mounted on a larger wooden base plate.

The vapor compressor is of the rotary type, as shown in the cross-sectional view, Fig. 1. The rotor, mounted eccentrically within the cylinder, has four radial slots containing blades which are thrown out by centrifugal force against the cylinder. Two pressure-equalizing slots are provided in the forward face of each blade. The various parts of the vapor compressor are also shown in the photograph, Fig. 5, taken after completing the test herein described.

The rotor shaft is supported in ball bearings mounted in the cylinder heads. The heads are bolted to the cylinder with

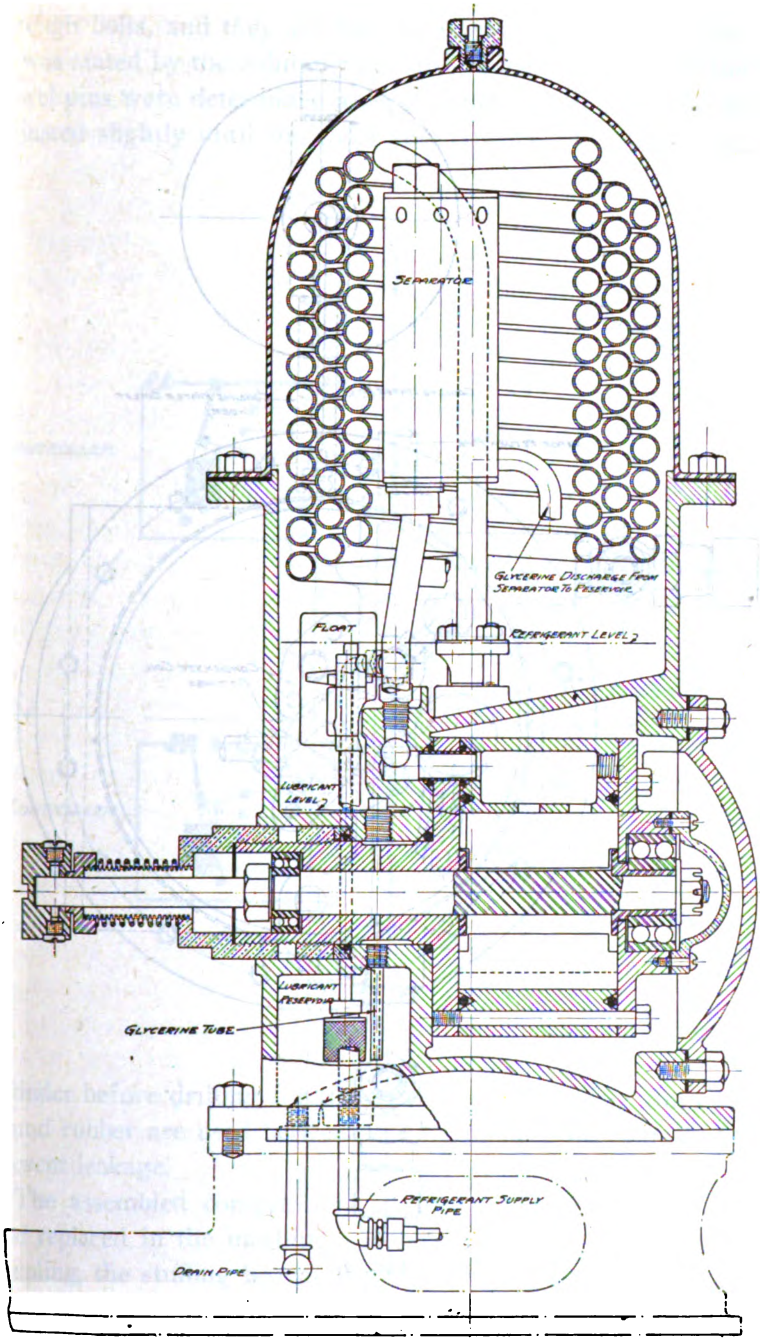

SEPARATOR
GLYCERINE DISCHARGE FROM SEPARATOR TO RESERVOIR
FLOAT
REFRIGERANT LEVEL
LUBRICANT LEVEL
LUBRICANT RESERVOIR
GLYCERINE TUBE
REFRIGERANT SUPPLY PIPE
DRAIN PIPE

FIG. 1.

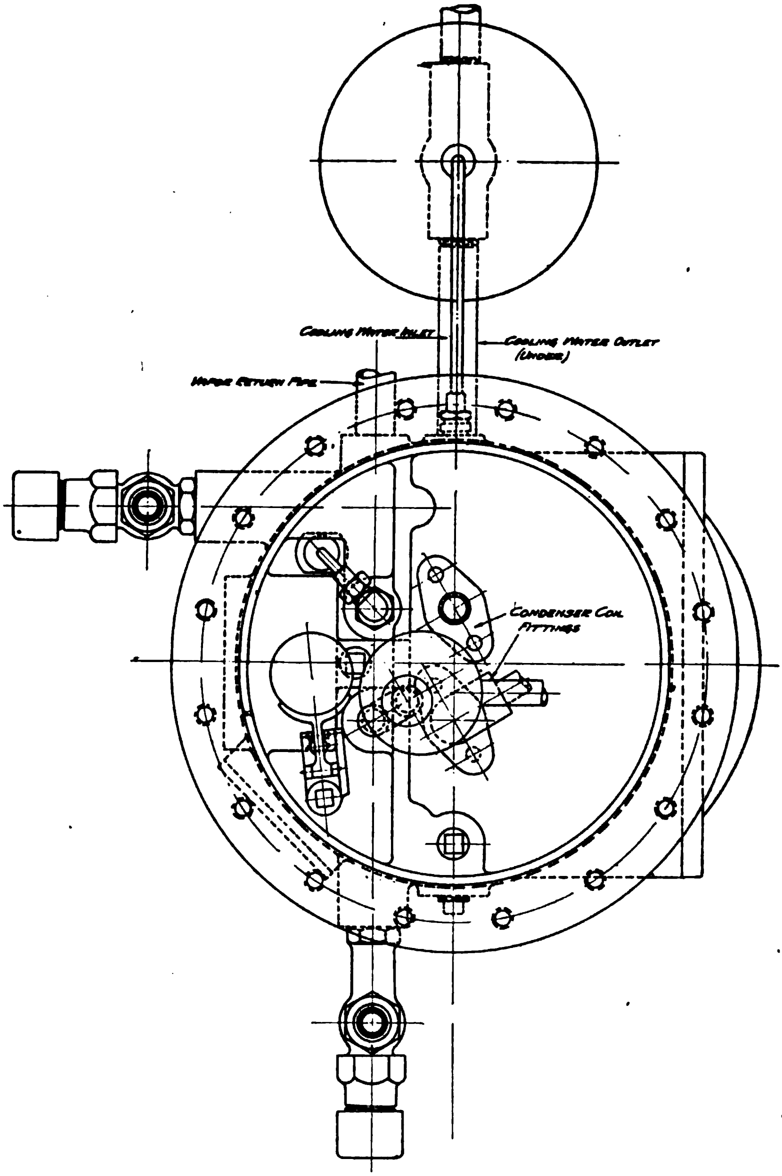

COOLING WATER INLET
COOLING WATER OUTLET (UNDER)
VAPOR RETURN PIPE
CONDENSER COIL FITTINGS

FIG. 2.

through bolts, and they are held in alignment by dowel pins. It was stated by the exhibitor that the proper locations of the dowel pins were determined after assembling, the heads being adjusted slightly until the rotor just touched the wall of the

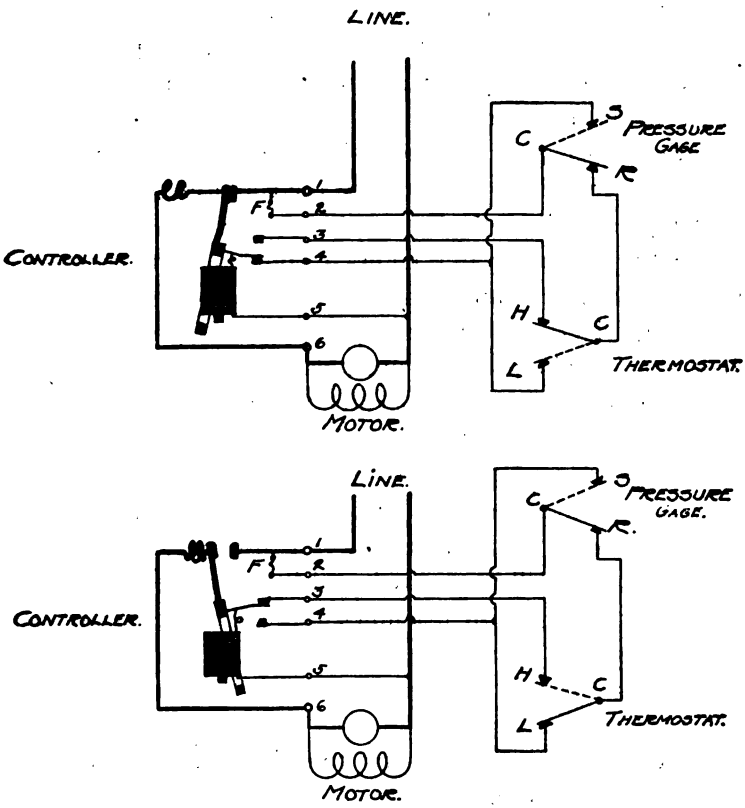


FIG. 3.

cylinder before drilling for the dowel pins. Packing rings of round rubber are used between the heads and the cylinder to prevent leakage.

The assembled compressor can be readily removed from and replaced in the machine. Referring to Fig. 1, the shaft coupling, the stuffing box and its nut are first removed. The

cover on the opposite side of the casing is then taken off and the assembled compressor withdrawn through the opening. The section and discharge connections to the compressor are made by two projecting nipples sealed into corresponding openings in the casing with round rubber packing rings. Similar packing rings are used to prevent leakage of cooling water between the compressor and the casing. It is proposed by the exhibitor to supply a spare compressor with each machine to be used on shipboard.

In the upper part of the casing, Fig. 1, are located the cooling-water coils for condensing the compressed vapor from the compressor below. On machines intended for domestic service the cooling water enters through a strainer and an automatic valve containing a diaphragm acted on by the compressed refrigerant. When the pressure of the refrigerant exceeds a certain predetermined value the valve is opened, permitting cooling water to enter the machine. When the machine is shut down the reduced pressure of the refrigerant permits a spring acting on the opposite side of the diaphragm to close the valve and thus automatically shut off the supply of cooling water. On machines for use on shipboard this automatic valve will be omitted, and it was therefore removed during this test. With machines for use on shipboard, it is also proposed to provide a small cooling water pump driven by an electric motor under the control of the same automatic devices as the main motor.

The cooling water enters by the pipe marked and passes through a hole in the casting to the nearer flanged fitting. The ends of the copper condensing coil are connected to the flanged fittings as marked. After flowing through the condensing coil the cooling water passes through a second hole in the casting and an opening under the plug shown into the space below surrounding the compressor. From this space the water is discharged through the outlet pipe below the inlet pipe.

The refrigerant vapor returns from an expansion coil or cylinder, located where the refrigeration is desired, through a strainer and a check valve to the pipe marked "vapor return" in Fig. 2, and thence directly to the suction side of the rotary compressor. From the discharge side of the compressor, the compressed vapor passes upward to the separator marked in Fig. 1 in which the entrained lubricant is separated from the vapor. The vapor escapes through the holes at the top of the separator and is condensed on the helical copper coils. When the liquid refrigerant level in the casing rises, the "float" is raised, thereby opening the valve to which the float is attached and permitting the liquid refrigerant to trickle through the pipe to the expansion coil or cylinder. A strainer is provided ahead of the float valve.

Lubrication is effected by the use of glycerine contained in the "lubricant reservoir." The "glycerine tube" is connected at its upper end to the vapor return pipe. A small orifice at the lower end of the tube, inclosed by a fine mesh strainer, determines the amount of glycerine that will be fed into the compressor. After lubricating the inner parts of the compressor, the glycerine is discharged with the compressed vapor into the separator, where it is separated from the vapor and returned through a drain tube to the lubricant reservoir. Lubrication of the ball bearings of the compressor is also insured by glycerine passing up the tube to the annular space shown, thence to and along the shaft to the ball bearing on the stuffing-box end. A hole through the compressor rotor conveys the glycerine to the ball bearing on the other end of the shaft.

A specially-designed stuffing box is used. The shaft coupling of case-hardened steel, pinned to the end of the shaft and revolving with it, is counterbored about ⅛ inch around the shaft. A projection on a steel cap is ground to fit into the counterbore of the coupling. This cap is held stationary as the shaft revolves by being connected by riflex tubing to the stationary bonnet. A helical phosphor-bronze spring presses

the cap into the coupling, but the pressure of the glycerine within the folds of the riflex tubing is sufficient to produce a tight joint, and it is evident that the tightness of the joint increases with the tendency to produce leakage by reason of the greater internal pressure.

Sight glasses are provided in the cylindrical casing, as seen in Fig. 4, to show the levels of the refrigerant and the lubricant therein. The casing may be drained of both refrigerant and lubricant through the "drain pipe." Pressure gages are provided to indicate the refrigerant suction and discharge pressures respectively. The latter gage contains a special electrical contact-making device for shutting down the motor should the discharge pressure become excessive.

The refrigerating machine tested was size B–5 and was driven by a 220-volt electric motor at 1,100 r.p.m., operated in conjunction with a thermostat and a pressure gage equipped with an electric contact device.

The automatic operation will be explained by the aid of the diagrams in Fig. 3, the heavy lines showing the main circuit, and the light lines the control circuits. The controller consists essentially of a contact device in the main motor circuit, operated by an electro-magnet which either opens or closes the circuit when energized momentarily. The upper diagram shows the position of the controller when the motor is running, the contact in the thermostat being made on the high-temperature point H, and that in the pressure gage on the running point R. Should the temperature operating the thermostat fall until contact were made on the low-temperature point L, the auxiliary circuit through the electro-magnet on the controller would be closed momentarily; consequently, the main circuit would be opened and the motor and refrigerating machine stopped. The auxiliary circuit would also be closed should the pressure rise until contact were made in the pressure gage on the stopping point S. The auxiliary circuit is always broken on the heavy contacts in the controller and not on the smaller contacts in the pressure gage and thermostat.

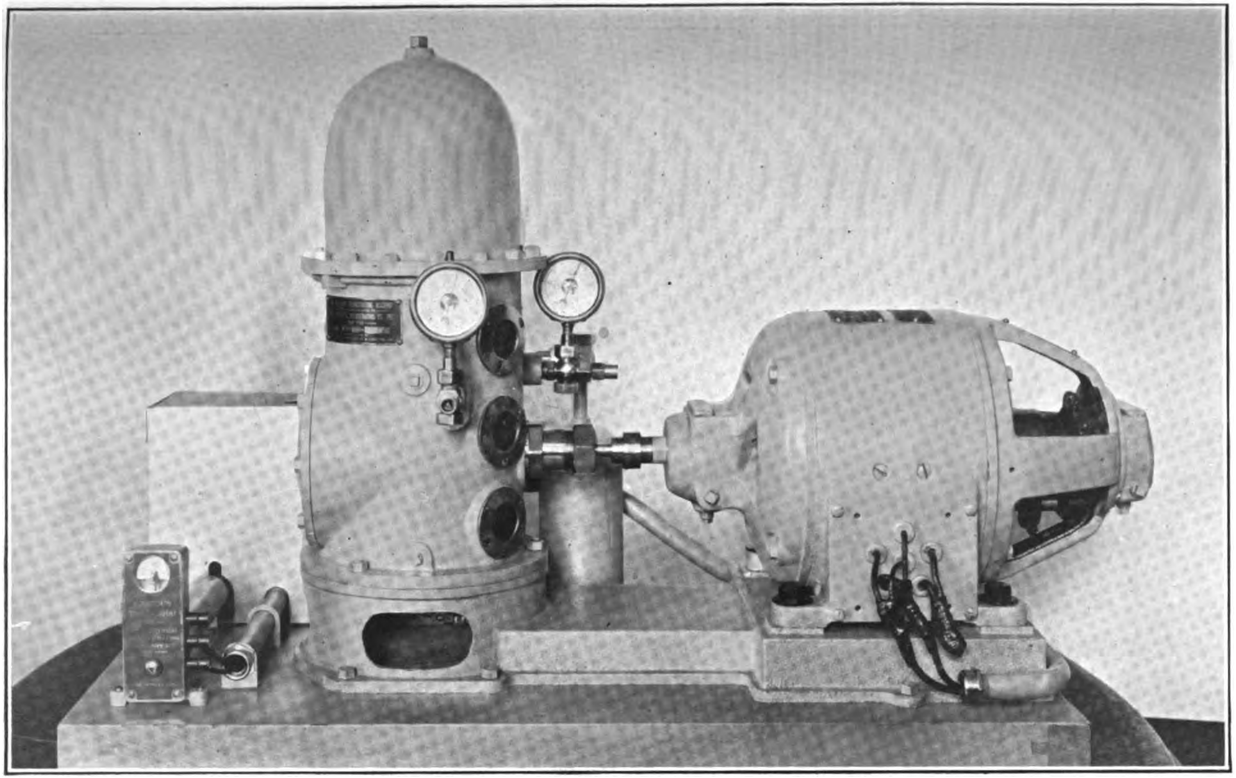


FIG. 4.

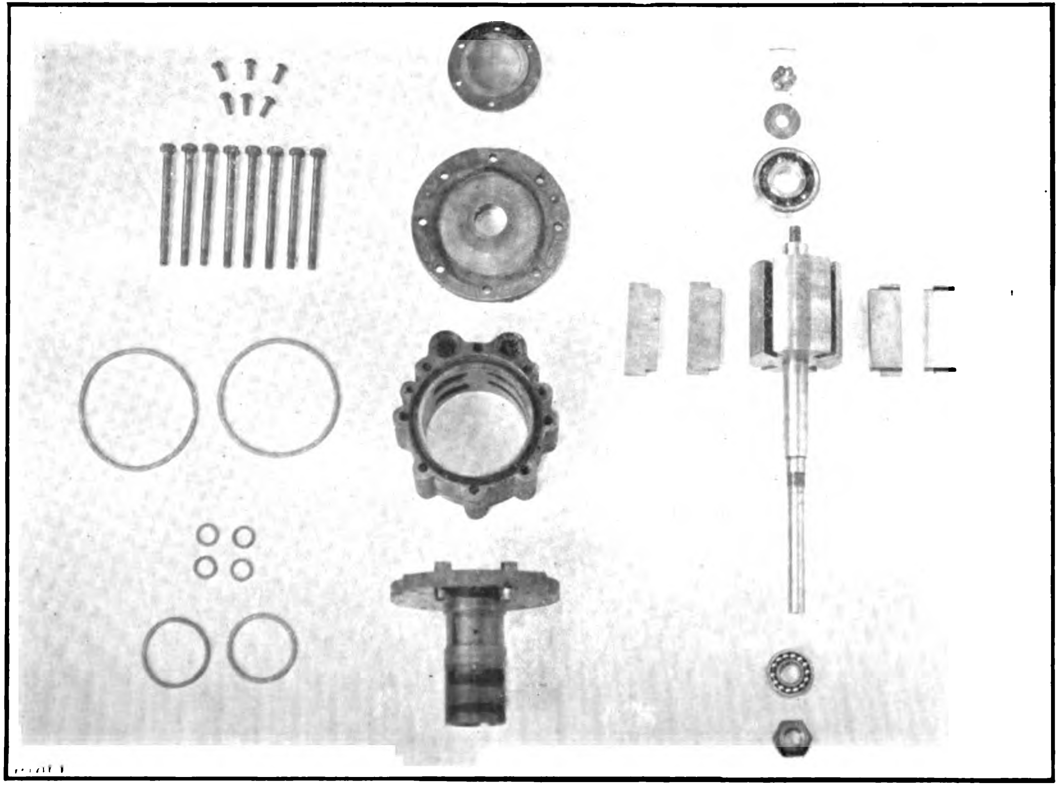


FIG. 5.

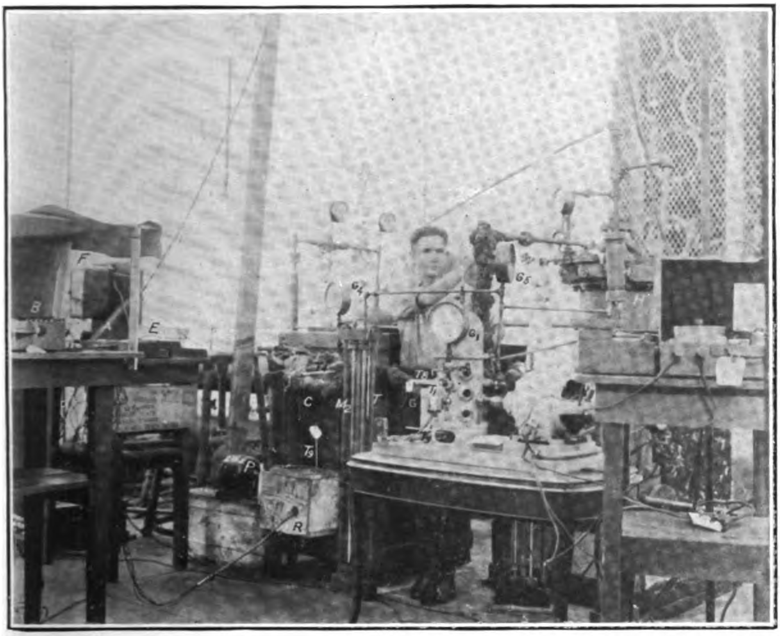


FIG. 6.

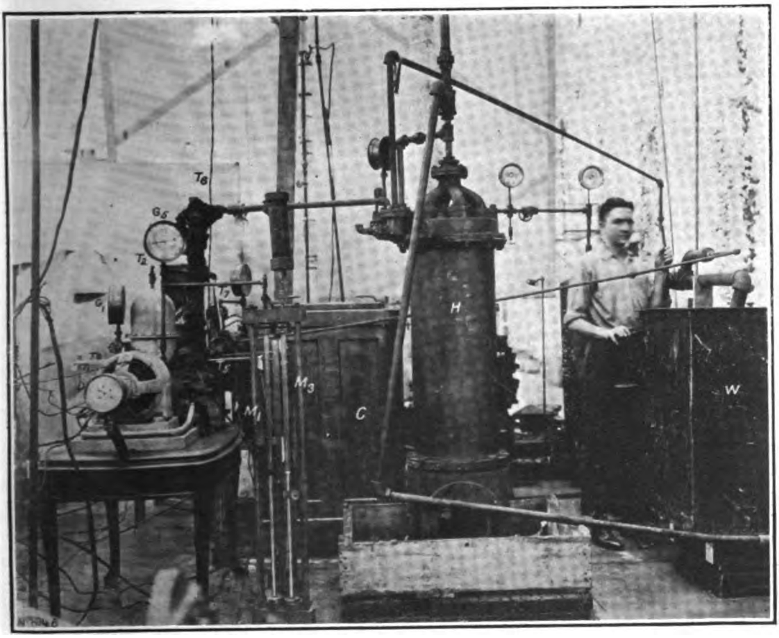


FIG. 7.

The lower diagram shows the conditions after the motor has been stopped due to low temperature. If the temperature rises the motor will start again; but should the motor stop by reason of rise in pressure it will not start again automatically when the pressure drops. A push button on the pressure gage must be depressed to again start the motor. A fuse F is provided in the auxiliary-control circuit.

The weight of the machine and motor with the automatic control apparatus mounted on a wooden base plate as shown was 264 pounds. The outside dimensions of the machine were 17½ inches wide by 37¾ inches long by 24½ inches high, equivalent to a volume of 9.3665 cubic feet.

With the refrigerating machine the exhibitor furnished an insulating cabinet, as seen in Fig. 6, containing a brine tank and an expansion cylinder to serve in conducting the test. The cabinet was about 29 inches square by 44 inches high. The brine tank within was 17 inches square by 33 inches deep. The expansion cylinder was 13 inches inside diameter, 14⅜ inches outside diameter and 26½ inches high, the refrigerant being contained in the annular space between the inner and outer walls. The liquid refrigerant supply line of ⅛-inch pipe was connected to the bottom of the expansion cylinder. The vapor line of ⅜-inch pipe branches and was connected to the top of the expansion cylinder at the opposite ends of a diameter. A ⅛-inch glycerin-return pipe, having a valve in it, was also connected from the bottom of the expansion cylinder to the vapor-return line.

OBJECT OF TEST.

The object of the test was to determine the refrigerating capacity and the economy of the machine under various temperatures of refrigeration. Also, to note the general operating characteristics of the machine and the durability of its parts.

TEST ARRANGEMENT AND CONDUCT.

The test arrangement is shown in the photographs, Fig. 6 and Fig. 7.

The electrical input to the motor driving the refrigerating machine was measured by means of an ammeter in the armature circuit, an ammeter in the field circuit, a voltmeter connected across the armature and a voltmeter connected across the series and interpole fields, all mounted on the table seen at the right in Fig. 6. The dial counter D seen in Fig. 7 served to measure the rotational speed of both the motor and the vapor compressor of the refrigerating machine. During the performance runs the automatic temperature control was not in operation. The pressure gage G, however, was connected to the controller to automatically stop the motor should the vapor discharge pressure become excessive.

The test gage G_1 served to measure the pressure of the compressed vapor from the rotary compressor. A mercury thermometer T_1 was inserted in the gage connection to measure the temperature of the compressed vapor. The bulb of this thermometer was directly exposed to the vapor and therefore subject to error by reason of the vapor pressure on the bulb. The thermometer T_2 projected into the vapor space within the condenser and was exposed to radiation to the surrounding cooling coils as well as to the pressure of the compresesd vapor. The thermometer T_3 was tied to the outside of the tube conveying the liquid refrigerant to the expansion cylinder within the cabinet.

The pressure and temperature of the vapor returning from the expansion cylinder were measured by means of the gage G_4 and the thermometer T_4 respectively. The bulb of the latter was directly exposed to whatever vacuum the refrigerant vapor was under, and also to radiation of heat from the pipe walls. The vacuum at the compressor inlet was indicated by gage G_5 and also by mercury column M_1. A mercury trap was provided on the latter to prevent loss of mercury. A

mercury thermometer, T_5, was also inserted in the compressor inlet. The mercury column M_2 measured the drop in pressure in the return vapor line between the expansion cylinder and the vapor compressor, including a check valve and strainer at the refrigerating machine. The thermometer T indicated the room temperature.

The temperature indicated on thermometer T_6, that of the cooling water supplied the machine, was regulated to the desired value by the amount of steam admitted to the surface heater H. The cooling water passed through the coils within the heater. The steam was admitted to the bottom of the shell, which was filled with water, an overflow being provided at the top. In previous tests it had been found that the temperature of the circulating water could be maintained much more nearly constant in this manner than if the steam were admitted in the usual manner and the condensed steam drained off at the bottom. The outlet temperature of the cooling water was measured by thermometer T_7. The thermometer T_8 projected into the cooling water leaving the condenser and entering the jacket space. The pressure drop of the cooling water flowing through the refrigerating machine was indicated on mercury column M_3. The cooling water was discharged into the tanks W and weighed.

The brine in the tank within the cabinet C was circulated by means of the motor-driven pump P. The expansion cylinder within rested on the bottom of the brine tank. The brine from the pump was discharged through a pipe projecting downwards within the expansion cylinder nearly to its base. Rising within the expansion cylinder, the brine overflowed the upper edge and passed downwards over the outer surface of the cylinder to the brine-pump suction pipe connected near the bottom of the brine tank.

The temperature of the brine in the brine-pump suction pipe was measured by the mercury thermometer T_9 and also by a platinum resistance element R connected to the special resist-

ance bridge B, with which was used a highly sensitive galvanometer not seen in the photographs.

A heating coil was also immersed in the brine within the expansion cylinder. The electrical input to this heating coil was measured by the ammeter and voltmeter E. The voltmeter leads were connected to the heating circuit at the cabinet C in order that the instrument should indicate the potential drop within the cabinet only. The watt-hour meter F was also connected in the heating-coil circuit. The aneroid barometer A served to measure the atmospheric pressure.

After setting up the apparatus as described above, the losses in the electric motor were first determined. The armature resistance loss for various armature currents was found by reading the voltage drop across the brushes for armature currents from zero to above the maximum full-load current, the shunt and series fields being disconnected and the armature slowly rotated by hand. The stray power loss was calculated from the armature current and voltage drop at various rotational speeds, obtained by varying the impressed voltage or the field current slightly from the normal value, the motor shaft being disconnetced from the shaft of the vapor compressor.

The platinum resistance element used to measure the temperature of the brine in the cabinet was calibrated after being connected to the bridge, by measuring its resistance in melting ice and also in condensing steam within a special hysometer.

All thermometers used were calibrated before the runs. The pressure gages were calibrated before and after the runs, and at times between the runs. The platform scales were adjusted to read accurately. The electrical instruments had been calibrated before the runs. The resistance thermometer was calibrated as described.

Runs were made at constant speed and with cooling water at room temperature and at 90 degrees F. Runs were also made in which the refrigerating machine was shut down and the valves to the expansion cylinder closed, in order to cal-

culate the heat imparted by the brine pump and absorbed from the room. The results deduced from the test data are plotted in Figs. 9 to 11 inclusive.

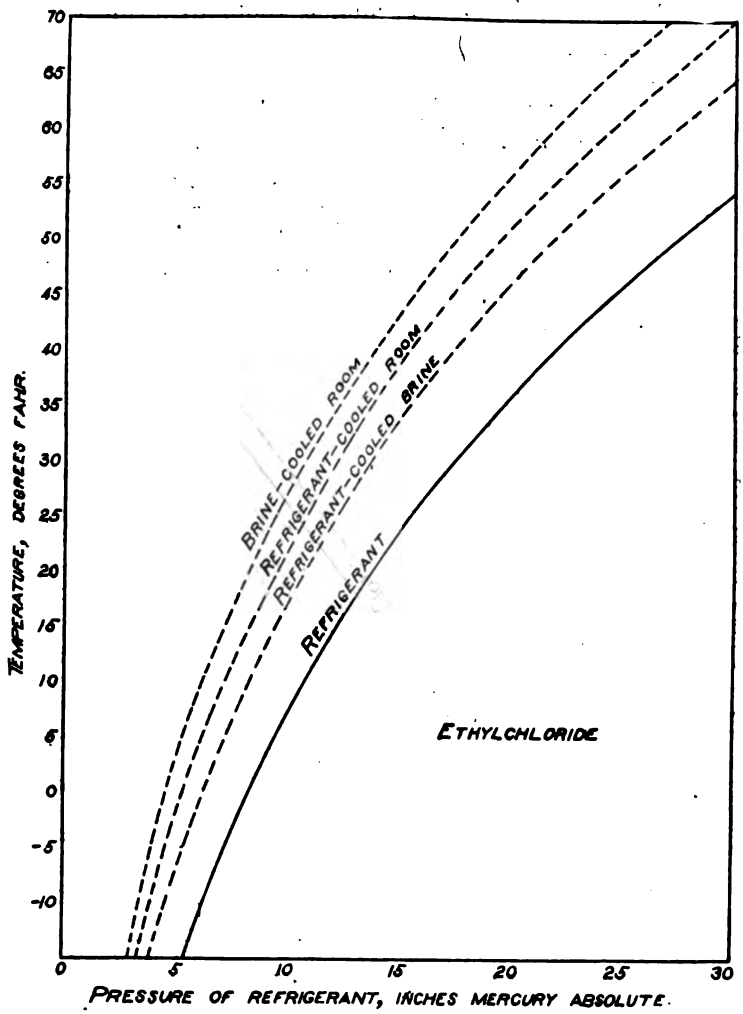


FIG. 8.

The saturated vapor temperatures of the refrigerant, ethylchloride, are taken from the full-line curve of Fig. 8, corresponding to the absolute pressures noted. In this figure are

also plotted dotted lines ten, fifteen and twenty degrees above the full-line curve. These temperature differences are considered fair values for cooling brine, for cooling a room by direct expansion, and for cooling a room by brine circula-

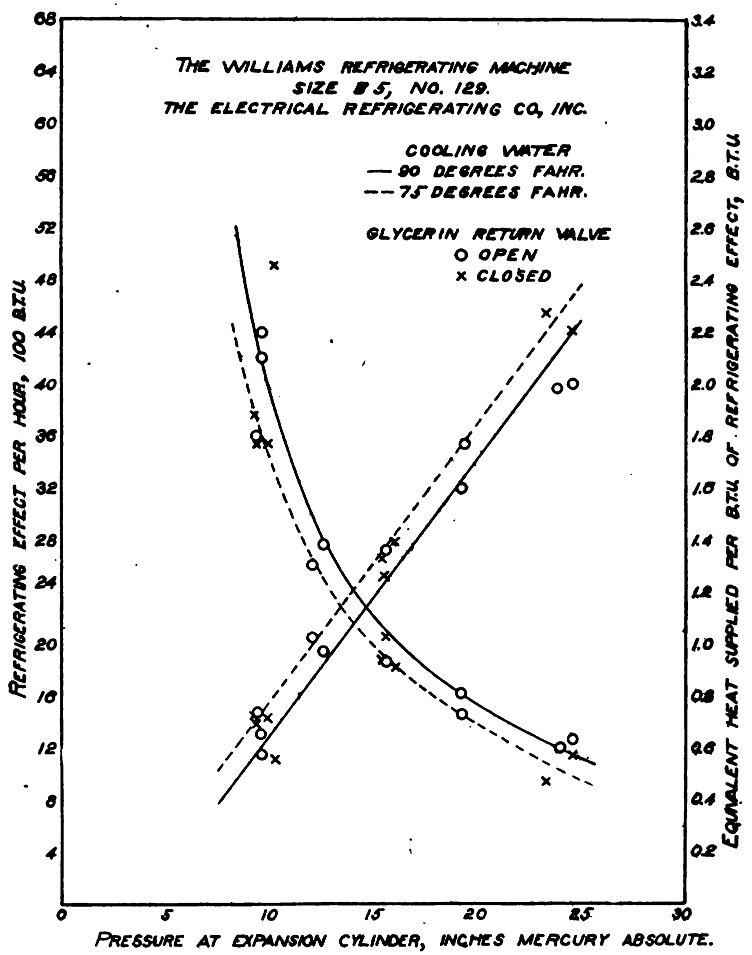


FIG. 9.

tion, respectively. In the case of a certain room which is to be kept at 40 degrees F., the vapor pressure carried in the expansion coils must be 13.8 inches mercury absolute if the room is cooled by brine circulation. If cooled by direct expansion,

the vapor pressure must be 15.4 inches mercury absolute. To cool brine to 40 degrees F., the vapor pressure would be 17.5 inches mercury absolute. More generous provision of sur-

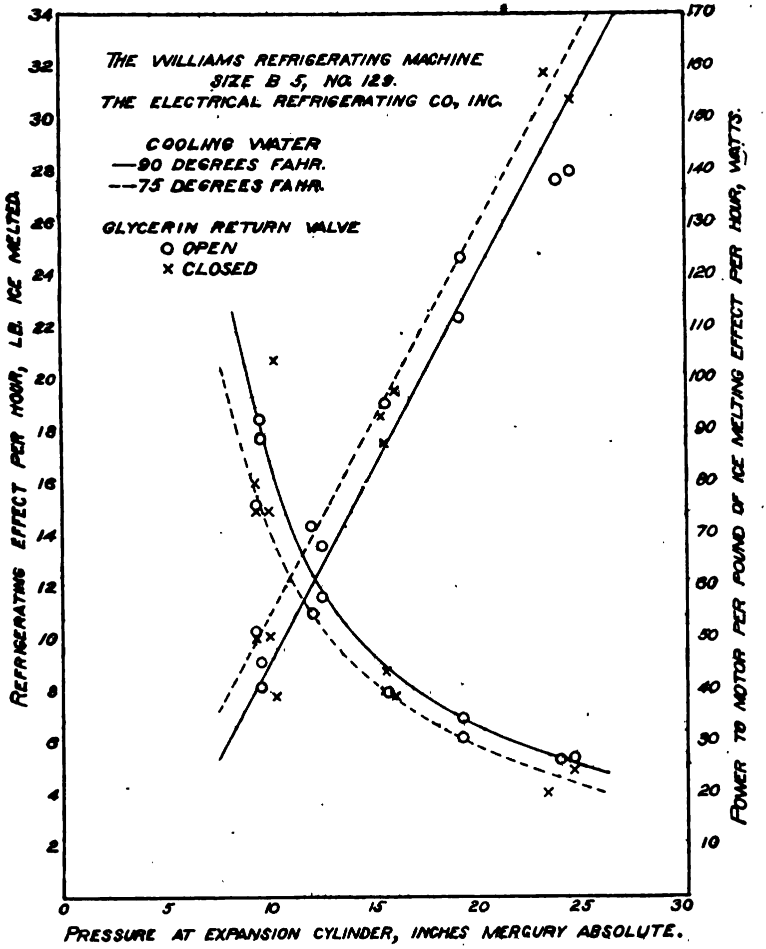


FIG. 10.

face in the expansion coils would, of course, result in lower temperature differences than indicated in Fig. 10. Thus, with the expansion cylinder furnished for this test, the temperature difference was only about five degrees F.

At whatever vapor pressure it may be necessary to operate

the machine tested, the corresponding capacity and economy are shown by the curves of Figs. 9 and 10. We find that with cooling water at 90 degrees F., the refrigerating capacities of this machine, corresponding to the absolute vapor pressures in the preceding paragraph, are 20.2 pounds of ice melted per hour to cool brine to 40 degrees F., 17.2 pounds per hour to cool a room to the same temperature by direct expansion, and 14.6 pounds per hour to cool a room to that temperature by brine circulation. These capacities correspond to the temperature differences on Fig. 10. If the temperature difference for cooling brine were only five degrees as with the expansion cylinder furnished, the vapor pressure for a brine temperature of 40 degrees F. would be 19.6 inches mercury absolute and the corresponding capacity would be 23.2 pounds of ice melted per hour. The capacity of this refrigerating machine when producing a given refrigerating temperature thus depends upon the extent and effectiveness of the surface in the expansion coils as well as upon the size of the machine.

The capacity of the machine is a function of the vapor suction pressure, as shown by the curves of inclosures (S) to (Z), inclusive. For a given temperature of refrigeration, the vapor pressure and consequently the capacity of the machine will depend upon the condition of operation. Thus,

For a room cooled by the circulation of brine, the capacities in Table 1 obtain:

TABLE I.

Temperature, degrees F.			Pressure of refrigerant, inches mercury absolute.	Capacity, ice-melting effect.	
Room	Brine	Refrigerant		Lbs. per hr.	Tons per day
60	50	40	21.9	26.7	0.321
50	40	30	17.5	20.2	0.242
40	30	20	13.8	14.6	0.175
30	20	10	10.8	10.2	0.122
20	10	0	8.3	6.4	0.077

Cooling water, 90 degrees F.

For a room cooled by direct expansion, the capacities corresponding to various temperatures are given in Table 2.

TABLE 2.

Temperature, degrees F.		Pressure of refrigerant, inches mercury absolute.	Capacity, ice-melting effect.	
Room	Refrigerant		Lbs. per hr.	Tons per day
60	45	24.5	30.6	0.367
50	35	19.6	23.2	0.278
40	25	15.4	17.0	0.204
30	15	12.2	12.2	0.146
20	5	9.4	8.1	0 097

Cooling water, 90 degrees F.

PROBLEM CORNER.

As an additional feature of the JOURNAL, friends of the Society have suggested that a few pages be devoted to the proposition of engineering problems and their solutions, the object being to stimulate interest in reviewing mathematical subjects and to establish a clearing house for the solution of engineering problems encountered in practice.

Such a feature, if successful, will be of value to the Society and the JOURNAL, in that it will provide a means for proposing problems whose solutions are of general interest and may well be the premises for discussion at length in the form of contributed articles.

To be successful this feature must arouse the interest and appreciation of the members of the Society and the subscribers to the JOURNAL. The principles involved in the problems proposed must be varied and the solutions must range in difficulty from simplicity to the latest methods of higher analysis developed for the treatment of practical problems in radio-telegraphy, elastic structures and thermodynamics.

Comment and criticism from the readers of the JOURNAL will be appreciated. If the problems proposed in this issue of the JOURNAL and the idea of incorporating similar problems in future issues meet with favor, effort will be made to publish as interesting and out-of-the-ordinary problems as can be obtained. Readers are invited to submit problems in which they are interested. Solutions will be published if of general interest and space permits.

PROBLEMS FOR SOLUTION.

Problem No. 1—Administration—An Actual War Problem.

In October, 1918, the Bureau of Steam Engineering received the following despatch from Admiral Sims:

"U. S. S. ———, after discharging completed, will require docking to renew broken stern bush. In order to reduce time

in dock, request you wire dimensions of brass stern bush as fitted in stern tube in order rough casting may be made before vessel docks. This information can not be obtained from vessel prior to docking."

The vessel referred to had been built on the Pacific Coast, and the only blue print showing dimensions of the stern-tube bushing was in the files of the Emergency Fleet Corporation in Philadelphia. A description of the stern-tube bushing with all dimensions was obtained by telephone and the following despatch transmitted:

"Outboard stern-tube bushing sixty and one-half inches overall comma outside diameter nineteen and one-quarter comma inside diameter eighteen comma outboard flange diameter twenty-three comma thickness one and one-quarter comma inboard keeper flange sixteen and one-half inside diameter thickness one-half period Inboard bushing twenty-four and one-half inches overall comma all diameters same as outboard bushing comma bulkhead flange twenty-three diameter thickness one and one-eighth comma set back three-quarters from inboard end period wood one inch thick period Shaft diameter over sleeve fifteen and fifteen-sixteenths comma bearing sleeve sixty-three and one-quarter inches long period All dimensions finished."

Referring to this case some time later, the Bureau was advised that "the cable in question came without error; the rough bushing was cast from the dimensions given, but not machined until the vessel was docked and finished dimensions checked. The results are very gratifying in that the casting was available on the day the ship docked, thus reducing the stay of the vessel in dry dock by four days."

Make a working sketch of the stern-tube bushings from which the foundry and machine shop work can be done.

Problem No. 2—Design—Mechanics—Theory of Elasticity.

In heavily loaded frame structures the elongation or compression of the various elements produces secondary stresses

in the elements which may be sufficiently great to cause failure of the entire structure—the Quebec Bridge disaster was claimed to have been so caused. To safeguard frame structures architects and civil engineers make use of the "principle of least work" in assigning safe stresses to the structural elements. The "principle of least work" states that the sum of the elements of work done in distorting the individual structural elements is a minimum for any given load on the entire structure. In reduction gearing for marine turbines the pinions transmit a considerable torque and undoubtedly suffer some torsional deflection, but for each half of each pinion to transmit the same tooth pressure there must be no relative torsional deflection between the two halves of the pinions. The Melville-McAlpine reduction gear recognizes this principle and provides for axial deflection of the floating frame.

Can the "principle of least work" be utilized in the design of pinion shafts or couplings so that the tooth pressures on each half of the pinion will always be equal? How?

Problem No. 3—Design—Mechanics—Calculus—Shop Practice.

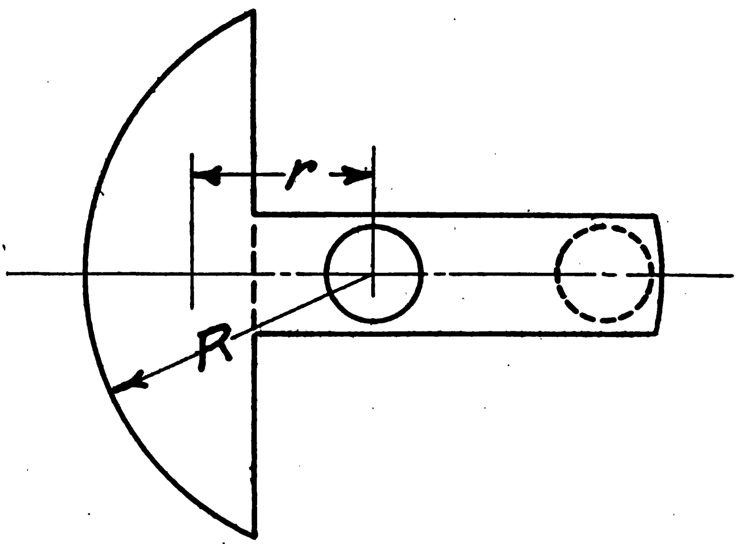


FIG. 1.

A designer, in laying out a counterweight for a crank shaft, plans to give the counterweight the shape of a circular segment the outside radius of which, R, is given. How can such a counterweight be designed so that its axial width may be minimum? This means, of course, that the product of the area of the segment by the distance, r, of its center of gravity from the center must be maximum. Having found the solution, indicate also the curve, giving the efficacy of such a counterweight between the two extremes, $r = R$ and $r = 0$, the result is perfectly simple and scientifically correct, but the designer does not like it and decides to sacrifice some of the features to secure a result more attractive from the standpoint of the practical engineer. By what considerations will he be guided in the contemplated departure from the exact solution?

Problem No. 4—Mechanics—Calculus.

A rod, 4.9 feet long, suspended by one of its ends, will perform small oscillations at the rate of 60 single beats per minute. It is also a well-known fact that, if suspended by a point, one-third the length below the top end, it will oscillate with the same frequency; while, when suspended at its middle, such a pendulum will, of course, have zero frequency.

Consequently there must be an intermediate point, such that the corresponding rate will be a maximum.

Required: (a) to locate such a point;
(b) to find the corresponding rate;
(c) to draw a curve connecting the rate with the corresponding point of suspension.

(This is understood to have been proposed originally by Admiral G. H. Burd.)

Problem No. 5—Mechanics—Calculus.

Find the strongest beam of rectangular section that can be sawed out of a round log whose diameter is D. This, of course, is a well-known problem, but it can be solved in two ways; one involves trigonometry and the other does not. Both can be obtained in two or three lines; having secured these,

draw a curve, of which the solution just found will be the maximum, and whose extremes correspond to the zero-breadth and zero-height conditions.

Problem No. 6—Mechanics—Calculus.

Find the stiffest beam of rectangular section that can be sawed out of a round log whose diameter is D.

Problem No. 7—Mechanics—Calculus.

Imagine a coil spring of the following characteristics: the extension of one inch means the pull of λ pounds; then λ is the so-called spring constant. To draw a diagram of such a spring assume that one end is fastened; take the other end as origin of coördinates and direct the axis of abscissae along the center line of the spring; the ordinates will represent the pull, and the deflection curve will, of course, be given by a straight line, beginning at the origin and inclined to the axis of abscissae at an angle such that its tangent λ.

It is required:

(a) To complete the diagram for the case when the spring is to work in compression as well as in tension.

(b) To draw a diagram for the case where a flat plate is placed between two springs, originally free, of characteristics λ and μ, respectively.

(c) Same as under (b), except that the springs are initially loaded, so that the flat plate is subject to a compression of p pounds on each side.

(d) Same as (b), except that inside each spring are placed shorter springs, of characteristics λ and μ, respectively; the latter springs do not come into play for deflections smaller than m on the right and n on the left side.

(e) Same as (c) and (d) combined.

Problem No. 8—Mechanics—Calculus.

Find general formulae of work (or energy), W, in terms of displacement, e, corresponding to the five types given in problem No. 7.

Find the displacement rate of energy dw/dl, and study the law of increase of the energy if the independent variable, l, is equicrescent.

Find the energy rate of displacement, dl/dw, and show the law of increase of the displacement, in each case, if the independent variable, W, is equicrescent.

Problem No. 9—Mechanics—Calculus.

A weight, P pounds, is supported by a coil spring, free length, l, characteristic, λ; find the period of free oscillation of such a system.

Problem No. 10—Mechanics—Calculus.

A weaker spring, length l_1, characteristic μ, is placed inside the spring of problem No. 9. Find the free period of the same weight, P.

Problem No. 11—Mechanics—Calculus.

If a half-inch hole is drilled through the shaft of a motor at 45 degrees to its axis, and if the end of a piece of cold-rolled steel 12 inches long is inserted and fastened therein, is such a system out of balance statically, dynamically or both? The system is started from rest and accelerated uniformly for t seconds, at the end of which period the speed is reached of n r.p.m., which thereafter remains constant. At what time is the rod in question subject to greatest stress, and what is the horsepower required in bringing the motor to that speed?

Problem No. 12—For Amusement—Algebra.

While not an engineering problem, the following is of historical interest:

"The combined ages of Mary and Ann are forty-four years, and Mary is twice as old as Ann was when Mary was one-half as old as Ann will be when Ann is three times as old as Mary was when Mary was three times as old as Ann. How old is Ann?"

NOTES.

NAVAL CONSTRUCTION (BRITISH) DURING THE WAR.

By Sir Eustace Tennyson d'Eyncourt, K.C.B., Vice-President.*

Having obtained the sanction of the Admiralty to give some particulars of vessels added to the fleet during the war, I suggested to the Council of the Institution when putting forward proposals for reading a paper, that in view of the very large amount of design work and construction carried out by the Admiralty during the period under review, it would, on the whole, be most useful and interesting to give a sketch and general summary of all the work rather than select a few types of warships and give a more detailed account of their design and construction.

I need hardly say that within the limits of time and space available for a paper read at this meeting it would be impossible to go fully into more than one or two of the most important designs, although perhaps it may be thought that a complete story of some of them, such as the *Renown* and *Repulse*, or the monitors, might make more attractive reading to our members. Still as the accounts of so many other important types would have had to stand over it appeared that the best course would be, as I have said, to give a general account of as many as possible of the ships added to the fleet during the last four and a half years, and this I will endeavor to do.

Immediately after war was declared, great pressure was exercised to complete the ships then building for the Navy, and to order such other vessels as could be designed and finished in the shortest possible time. Many very wise people told us the war would only last a year or possibly eighteen months—in fact that it could not be carried on for a longer period.

This view necessarily colored all that was done in the way of naval design and construction. Generally speaking, therefore, the construction of new battleships was ruled out. With the acquistion of the *Agincourt*, *Erin* and *Canada*, which were building here for foreign Governments in private yards, and bearing in mind the early completion of the remaining two vessels of the "*Iron Duke*" class, shortly to be followed by the vessels of the "*Queen Elizabeth*" class, we had a great preponderance of heavier capital ships, or Dreadnaughts, over the enemy, and as this class of ship takes longer to design and construct than any other, it was obviously a prudent course to concentrate on such types as were specially needed and could be built more quickly.

It should also be remembered that the menace of the submarine, which was from the first beginning to loom as a vital factor in the war, pointed in the direction of large numbers of patrol boats, torpedo-boat destroyers, and smaller types of vessels to deal with this menace. No time, therefore, was lost in placing orders for additional destroyers, submarines, light cruisers, sloops, mine-sweepers, patrol boats, &c.; and it very soon became clear that the dockyards and the regular warship-building contractors would not be able to cope with the mass of new construction that was required.

Accordingly, orders for many of the last-named classes were placed with builders who had hitherto only been accustomed to mercantile work.

* Paper read before the Institution of Naval Architects, April 9, 1919.

With the arrangements that were made, however, for superintending and overseeing the work by the Admiralty, with the assistance of the registration societies—Lloyd's and the British Corporation—very little difficulty was experienced in getting the work satisfactorily carried out by the firms new to this class of shipbuilding, and I think the results show what success attended the arrangements made.

To take ships added to the Navy during the war in the proper order, it is necessary to begin with battleships of the "*Iron Duke*" class. The particulars of all previous Dreadnaughts are pretty well known, and have been published. The "*Iron Duke*" class, of which there were four, followed the "*King George V*" class both in sequence of time and in general characteristics. The same main armament, similarly arranged, with the five turrets all on the center line of the ship, was adhered to, the chief difference in the "*Iron Dukes*" being that instead of the 4-inch guns forming the secondary armament, a battery of twelve 6-inch guns protected by 6-inch armor was, after considerable discussion, finally decided upon. The protection also was somewhat increased over that of the *King George V*, involving an increase in dimensions over any of our previous battleships, due to the addition of these weights and of other items. Two of the class had been laid down in January, 1912, and two in May, the four vessels being completed in March, June, October and November, 1914, so that two were ready just before, and two shortly after, the declaration of war. Four torpedo tubes were carried in lieu of three in the previous ships, and after the Battle of Jutland, a considerable amount of additional protection was added over the magazines—a course which was practically adopted in all our ships at that time as a precautionary measure. Only in one case was any portion of a shell found to have penetrated below the protective deck; but with the ever-increasing range at which actions have been fought, and the increasing penetration of improved shell, the danger of the decks being inadequate had to be considered.

The tables herewith give general particulars of these ships and of all the others with which I am dealing in this paper. It should be mentioned, however, that the speed obtained on trial was approximately 22 knots, or about a knot in excess of the legend speed.

Special interest is attached to this class, as the *Iron Duke* was the fleet flagship during the whole time of Admiral Jellicoe's appointment as Commander-in-Chief, and she was in action at Jutland with her sister ships. The *Marlborough*, it should be specially noted, was the only British battleship of the post-Dreadnaught type struck by a torpedo during the whole war, and the value of the longitudinal protective bulkhead and of the subdivision and arrangements adopted was clearly shown, as the ship was able to remain in the line, no vital damage being done. She was afterwards safely docked in the Tyne and repaired. This is, I think, specially interesting, as many of our older ships, some with center-line bulkheads and with other arrangements not so good for dealing with under-water damage, were sunk in the Dardanelles and elsewhere by enemy torpedoes.

The next type to note is the "*Queen Elizabeth*" class of the 1912-13 program. Three of these vessels, after taking a little more than two years to build, were completed in January, March and October, 1915. The other two were completed in February, 1916. A very considerable departure was made in the *Queen Elizabeths* from any previous Dreadnaughts, the 15-inch gun taking the place of the 13.5-inch and the designed speed being increased by 4 knots over our previous Dreadnaughts, whilst the secondary armament was similar to that of the *Iron Dukes*, consisting of 6-inch guns. Their very great increase of speed-involved practically doubling the horsepower necessary to give the 25 knots desired, and the

great increase in the weight of the 15-inch guns and mountings over the 13.5-inch meant accepting only four turrets with eight 15-inch guns, as against five turrets with ten 13.5-inch guns in our previous ships, and even so, the armament was considerably heavier. The further great departure from previous practice in battleships was the adoption of oil only, as the fuel. This necessitated special arrangements of the oil bunkers, many of which were 30 feet in height, and required special construction to withstand the head of oil. The armor and protection were fully maintained as compared with previous ships, but all these additions involved increasing the displacement of 27,500 tons.

In the Battle of Jutland, the Fifth Battle Squadron, consisting of four vessels of this class, were heavily engaged for several hours, and although they inflicted and sustained heavy punishment, especially in the case of the *Warspite,* all the vessels gave a splendid account of themselves and were not seriously damaged or put out of action. After the Battle of Jutland, additional protection was added to the magazines. It may be mentioned that the oil fuel proved a complete success, it being found easier to keep up a high sustained speed, and a smaller complement also is, of course, involved, as there is great saving in *personnel* as against that required for a coal ship.

I should mention that Sir Philip Watts was responsible for the design of the "*Iron Duke*" and "*Queen Elizabeth*" classes, thus completing a series of 27 battleships of the Dreadnaught class designed and built during his tenure of office at the Admiralty—in addition to the large number of battle cruisers and light cruisers and other vessels built during that period—truly a great record.

BATTLESHIPS OF THE "ROYAL SOVEREIGN" CLASS.

The next in order came the "*Royal Sovereign*" class of the 1913-14 program. These vessels were to have the same armament as the *Queen Elizabeth,* but as there was some question about the supply of oil fuel when the design was discussed, it was decided to revert to coal, and also to accept the slower speed of 21 knots, which would make them more homogeneous with other Dreadnaughts. Subsequently, when the vessels were in process of construction and the great advantages of the use of oil fuel with other types of warships became apparent, it was decided to change from coal to oil, and it was anticipated that increased power giving a speed of about 23 knots would be obtained. As a matter of fact when fully laden with about 4,000 tons of oil, the *Revenge* attained 22 knots, which was equal to about 23 knots in the designed load condition. A different disposition of deck and side armor was also adopted by which the thick protective deck at the center of the ship was brought up to the level of the main deck; this portion of the protective deck being thus well above the level of the deep load-line, and giving more protected freeboard in the damaged condition than in any of our earlier battleships. This was an important feature, a somewhat reduced metacentric height was decided upon for these ships with a view to making them steadier gun platforms than some of the ships with larger G.M. The vessel was provided with good under-water protection, which in certain of the ships was further reinforced by adding outside bulge protection. This was done to the *Ramillies* before her launch and also to two other vessels of the class after they had been in commission some time, during refit, and it is proposed to add the bulge to the remaining two ships of the class when opportunity offers. The addition of bulges was suggested by myself originally for the "*Edgar*" class for which I designed this form of protection in 1914, after considerable experiments had been made. The results have proved the efficiency of the bulges. Further particulars of the "*Royal Sovereign*" class can be seen from Fig. 1 and the table.

BATTLESHIPS TAKEN OVER FROM FOREIGN GOVERNMENTS.

The three ships taken over from foreign Governments were of different types, as shown in the table of particulars and Figs. 2 and 3.

H. M. S. *Agincourt* (Fig. 2) was commenced in September, 1911, for the Brazilian Government, when I took out various designs (got out under Mr. Perrett, at Elswick) to Rio de Janeiro, and finally settled on the design of the *Agincourt,* after modifying it very considerably on the spot. The Brazilian authorities, after much discussion, decided upon fourteen 12-inch guns, twin-mounted in seven turrets. This involved a ship with a length of 632 feet between perpendiculars and 671 feet 6 inches over all. The main armor was somewhat lighter than our British Dreadnaughts (as seen from the particulars opposite), and in other respects, such as fueling facilities, the ship hardly came up to the British standard. However, she was well reported on and the 14 big guns were liked by the gunnery officers, who preferred a large number of guns for their salvoes. Certain alterations had to be made to fit her for our service, but in the main she was left as designed. I should perhaps mention that in 1914 she was transferred from the Brazilian Government to the Turkish Government, and when war broke out she was on the point of leaving for Constantinople, when she was taken over.

The design of the Erin, Fig. 3, was settled by the three firms—Armstrongs, Vickers, and Brown—in consultation with the Turkish authorities, for whom the vessel was built, being commenced in November, 1911. In general characteristics she more nearly followed the "*King George V*" class than any other British ship, except that the secondary armament consisted of 6-inch guns, as in the "*Iron Duke*" class. This vessel also was taken over by the British Government in August, 1914, and certain modifications made to fit her for the British service. In respect of quantity of fuel carried, the *Erin* was below the standard adopted for vessels designed for the British Navy.

The third ship taken over from a foreign Government was ordered and commenced in 1911 at Elswick, from designs prepared by Mr. Perrett for the Chilean Government. There were two ships of the class, the *Almirante Latorre* (now the *Canada*), and the sister ship the *Almirante Cochrane* (now the *Eagle*). The *Canada,* Fig. 4, has ten 14-inch guns, twin-mounted, in the center line, and was originally designed to have twenty-two 4.7-inch as the secondary battery, but this was subsequently altered to sixteen 6-inch guns. The protection, again, was somewhat lighter than that of our Dreadnaughts, but the speed was rather higher, viz, 22¾ knots, and as a matter of fact this speed was considerably exceeded on trial. The ship was taken over by British Admiralty in September, 1914, and completed, after certain necessary modifications, a year later. Her fuel consisted of coal, with the addition of a certain amount of oil, as in most of our battleships. The sister ship, *Almirante Cochrane,* remained in an uncompleted condition on the stocks at Elswick till the spring of 1917, when she was taken over by the British Government and rearranged as an aircraft-carrying ship. She was renamed H. M. S. *Eagle,* and as a compliment to the United States Navy, she was, at the request of the Admiralty, launched by Mrs. Page, the wife of the late American Ambassador. The above record finishes the list of battleships proper which are completed for service during the war.

LARGE BATTLE-CRUISERS.

Coming to the battle-cruisers, particulars have already been published of all these down to H. M. S. *Tiger.* This ship was included in the 1911-12 program and followed on the *Queen Mary,* the general features of the

two ships being much alike, the chief differences being that the secondary armament of the *Tiger* is twelve 6-inch guns in lieu of sixteen 4-inch in *Queen Mary,* and *Tiger* has two submerged torpedo rooms, whereas the *Queen Mary* had only one.

After the design was approved by the Board, the ship was ordered and laid down at Clydebank on June 12, 1912, and completed in October, 1914. In common with so many of our ships completed during the war, the early commissioning and joining of the Fleet was so imperative, that no exhaustive trials in deep water were carried out, but the runs made on the Polperro Course showed that the designed power of 108,000 shaft horsepower could be obtained with little difficulty, corresponding to a speed of 30 knots. During the progress of the design, the oil-fuel capacity was very largely increased in case of need, the original tanks which only allowed for 1,100 tons were supplemented to admit of a maximum oil stowage of 3,480 tons, in addition to the 3,320 tons of coal; but it is not usual for the vessel to carry this full fuel stowage, at any rate, of oil.

BATTLE-CRUISERS "RENOWN" AND "REPULSE."

At the commencement of the war two additional battleships of slightly modified Royal Sovereign type, viz, the *Renown* and *Repulse,* had been laid down, but in view of the long time it would take to complete these ships, the construction was not pressed forward. Immediately after the battle of the Falkland Islands, in which our battle-cruisers *Invincible* and *Inflexible,* in company with other smaller cruisers, annihilated Von Spee's fleet, the value of the battle-cruiser type became very apparent, and on the initiative of Lord Fisher, then First Sea Lord, it was decided to stop the construction of the *Renown* and *Repulse* as battleships and to alter the design completely into that of very fast battle-cruisers. I received instructions to re-design these ships about Christmas, 1914. The new design had to give a speed of 32 knots, with the largest number of guns possible for such a vessel, and with protection similar to the *Invincible* and *Indefatigable* classes. A modified form of bulge was adopted in these ships to give additional underwater protection against torpedo attack. Proposals are now under way for still further adding to this bulge protection.

The general outline design was completed and approved in ten days, and six 15-inch guns were adopted as the main armament, the secondary armament consisting of seventeen 4-inch guns, of which fifteen were mounted in five specially-designed triple gun mountings. Owing to the circumstances, referred to above, it was necessary that the ships should be completed at the earliest possible date, and I suggested that the *Tiger's* machinery should be repeated with some additional boilers, and with the extra length it was found possible to obtain the speed of 32 knots, as laid down by the Board. Lord Fisher also insisted that the ships must be completed with fifteen months—an abnormally short time for an entirely new design, without any drawings prepared. This period of completion was not realized, although not greatly exceeded.

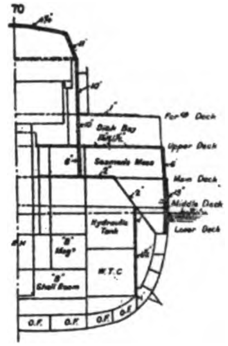
By January 21, 1915, the two firms entrusted with the orders, viz, John Brown and Co. and the Fairfield Company, were supplied with sufficient information to enable them to proceed with the structure, and both keels were laid on January 25. All the drawings and specifications were completed by April and the design finally approved in that month. The fuel was to be entirely oil, and with the additional boilers the power expected to be from 110,000 shaft horsepower to 120,000 shaft horsepower—the latter having actually been obtained on trial. The arrangement of the whole ship, showing the protection, is given in Fig. 5, the plating over the magazines being considerably increased during the construc-

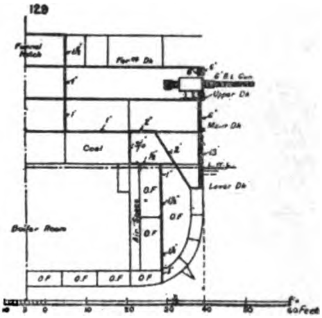
Speed at load draught (knots) submerged	10	10	10¼	9¼	[illegible]	10¼
Oil fuel capacity (tons)	45	44	16	91	200	76
Armament .. .. —	1 3-in. 5 18-in. T.T.	1 3-in. 4 18-in., 1 21-in. T.T.	— 4 21-in. T.T.	1 3-in. or 4-in. 6 18-in. T.T.	1 4-in., 1 3-in. H.A. 8 18-in. T.T.	1 3-in. or 4-in. 6 18-in. T.T.

TABLE I.—PARTICULARS OF BRITISH WARSHIPS CONSTRUCTED DURING THE WAR.

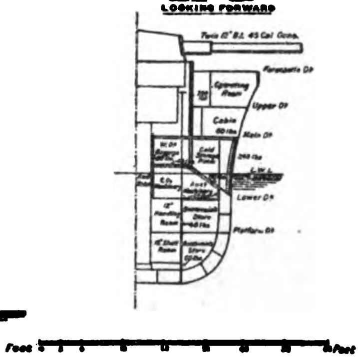
	Torpedo-Boat Destroyers.			T.B.D. Flotilla Leaders.		Patrol Boats.	Sloops and Mine-Sweepers.			"China" Gunboats.	
—	"M" Class.	"R" and "S" Classes.	"V" and "W" Classes.	"Kempenfelt" Class.	"Scott" Class and "Shakespeare" Class (approximately same)	"P" Class.	Single-screw Sloops, "Flower" Class.	Paddle Mine-sweepers.	Twin-screw Mine-sweepers.	"Fly" Class.	"Insect" Class.
Length between perpendiculars ..	265 ft. 0 in.	265 ft. 0 in.	300 ft. 0 in.	315 ft. 0 in.	320 ft 0 in.	230 ft. 0 in.	255 ft. 3 in.	235 ft. 0 in.	220 ft. 0 in.	120 ft. 0 in.	230 ft. 0 in.
Length overall	273 ft. 4 in.	276 ft. 0 in.	312 ft. 0 in.	325 ft. 0 in.	332 ft. 6 in.	244 ft. 6 in.	267 ft. 9 in.	245 ft. 9 in.	231 ft. 0 in.	126 ft. 0 in.	237 ft. 6 in.
Breadth, extreme	26 ft. 8 in.	26 ft. 8 in.	29 ft. 6 in.	31 ft. 9 in.	31 ft. 9 in.	23 ft. 9 in.	33 ft. 6 in.	29 ft. 0 in. (58 ft. over paddles)	28 ft. 0 in.	20 ft. 0 in.	36 ft. 0 in.
Load draught, mean	8 ft. 8 in.	9 ft. 0 in.	9 ft. 0 in.	10 ft. 0 in.	10 ft. 6 in.	7 ft. 7 in.	11 ft. 0 in.	6 ft. 9 in.	7 ft. 0 in.	2 ft. 0 in.	4 ft. 0 in.
Displacement in tons	1,025	1,065	1,300	1,650	1,800	573	1,250	810	750	98	645
Shaft horse-power of engines	25,000	27,000	27,000	36,000	40,000 to 44,000	4,000	2,400	1,400	1,800	175	2,000
Speed at load draught (knots)	34	36	34	34	36	22	17	15	16	10	14
Fuel at load draught (tons)	140	150	185	255	250	50	130	—	—	—	—
Coal capacity (tons)	—	—	—	—	—	—	260	150	140	5	35
Oil fuel capacity (tons)	280	300	370	515	500	96	—	—	—	10	54
Armament	3 4-in. 1 2-pdr. 4 21-in. T.T.	3 4-in. 1 2-pdr. 4 21-in. T.T.	4 4-in. or 4·7-in. 1 3-in. 4 or 6 21-in. T.T.	4 4-in. 2 2-pdrs. 4 21-in. T.T.	5 4·7-in. 1 3-in. H.A. 6 21-in. T.T.	1 4-in. 1 2-pdr. 2 14-in. T.T.	2 4-in. or 4·7-in. 2 3-pdrs. —	1 3-in. 1 6-pdr., 2 2-pdrs. —	1 3-in. 1 6-pdr., 2 2-pdrs. —	1 4-in. 1 3-in., 1 6-pdr. —	2 6-in. 2 3-in. —

SECTIONS LOOKING FORWARD

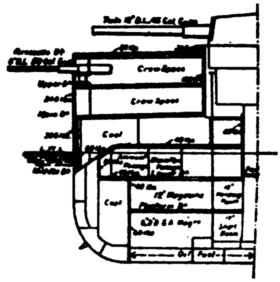




SECTION AT 69
LOOKING FORWARD



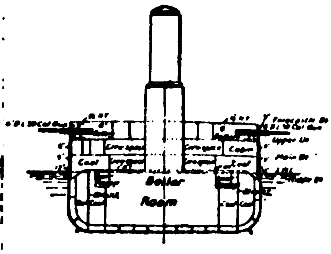
SECTION AT 179
LOOKING AFT



PRO

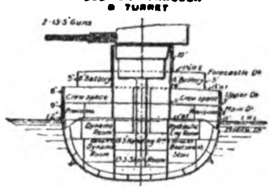
UPPER FORECAST

SECTION THROUGH
BOILER ROOM

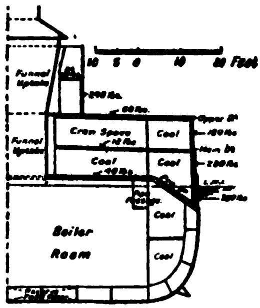


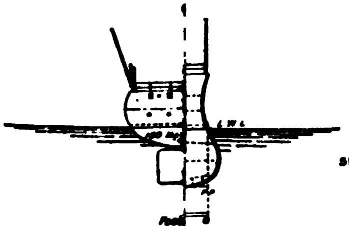
UPPER. FO

SECTION THROUGH
B TURRET



SECTION THRO' Nº 4 BOILER ROOM
LOOKING AFT.





SECTION THRO' A BARBETTE
LOOKING FORWARD.

UPPER, FORECASTLE & S

Digitized by Google

N THROUGH BOILER ROOM.

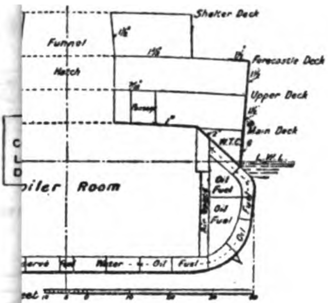
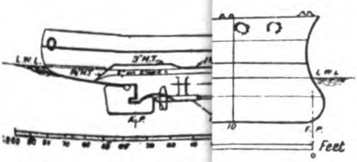


Fig. 6. "COURAGE

COMMENCED BUILD
LAUNCHED
COMPLETED

PROFILE



FORECASTLE AND SHEL

SECTION AT 96

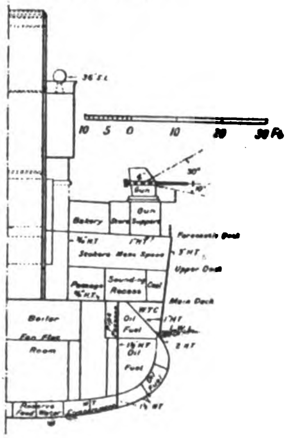
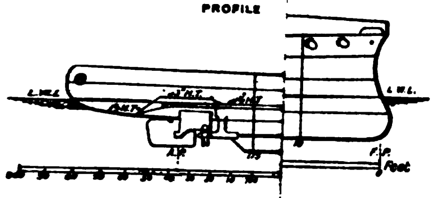


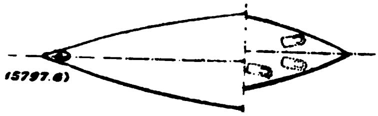
Fig. 7. FU

Commenced
Launched
Completed

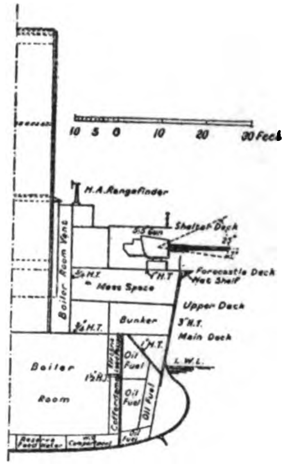
PROFILE



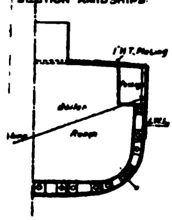
FORECASTLE AND SHELTER

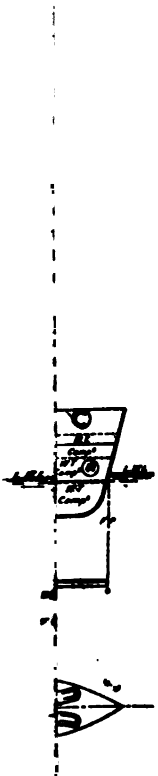


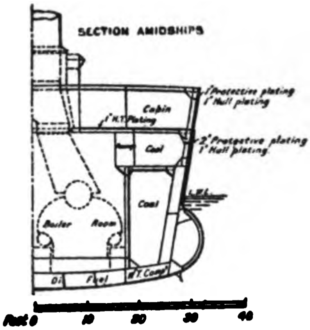
SECTION AT 96.



SECTION AMIDSHIPS.

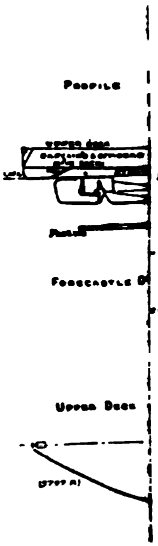


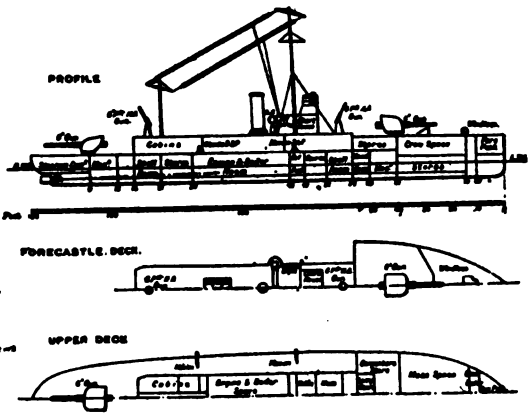




6" MONITORS. (M 29 to 33)

Fig. 10. MONI





tion as a result of the Jutland fight. The *Repulse* was launched in January, 1916, less than a year from the laying down, and the *Renown* was launched two months later. The *Repulse* went through her commissioning trials early in August, and the *Renown* followed one month later and was completed in September. The speed of *Repulse* on trial was over 31¼ knots in the deep condition, and the *Renown* obtained 32.6 knots mean speed on the new measured course off Arran in the normal condition. The ships have been well reported upon at sea and maintain their speed well. I think that the construction of these vessels in a little over one and a-half years from the first order to get out the design, constitutes a record in design and construction of two such important vessels, and reflects great credit, not only upon the Royal Corps of Naval Constructors, but also upon the contractors and all concerned in the construction and completion of the vessels. In fact, the Admiralty conveyed their appreciation of this to me in a letter dated September, 1916.

LARGE LIGHT CRUISERS "COURAGEOUS," "GLORIOUS" AND "FURIOUS" (FIG. 6 AND FIG. 7).

Whilst the designs of the *Renown* and *Repulse* were in progress, I received instructions to design some very high-speed ships carrying powerful guns and of a size sufficient to keep their speed in moderate weather, but to have a draught lighter than any existing British or enemy ship of the same class, so as to be able to navigate shallow waters, if required.

As sanction was not likely to be obtained for building more capital ships taking two years or longer to complete, while additional light cruisers had been already approved of, it was decided to build the *Courageous* and *Glorious* on the lines of very light cruisers mounting a few guns of heaviest caliber, so as to be able to annihilate any enemy light cruisers or raiders. They were to have thin protection, similar to our light cruisers, and a speed of not less than 32 knots, the draught being restricted to about 22 feet, or about 5 feet less than any existing battelship or battle-cruiser carrying such heavy guns, the main armament of four 15-inch guns in two turrets, one forward and one aft, making them a match for any raider or light cruiser that might be encountered. At this time it should also be remembered that the armaments of ships, especially as regards heavy guns, had to be regulated by the guns and gun mountings which would be available or could be manufactured in the time at our disposal, and this condition applied to the 15-inch mountings which were adopted for these ships. The secondary armament consisted of eighteen 4-inch guns in six triple mountings, similar to the triple mountings of the *Renown* and *Repulse*. The side armor consisted of 2-inch protected plating on top of the 1-inch shell plating, as in our light cruisers, and a thin protective deck was worked all fore and aft, but this was considerably thickened over the magazines after Jutland. A modified bulge was arranged for, as in the *Renown* and *Repulse*.

The machinery adopted for these ships was of the type fitted in the light cruiser *Champion*. It consisted of a four-shaft arrangement of geared turbines, the power being transmitted to the propeller shafts by double helical gearing. The 18 boilers of Yarrow small-tube type were also similar to those of the light cruisers, and with all-oil firing a power of 90,000 shaft horsepower at about 340 revolutions was aimed at. Such trials as it was possible to make showed that 32 knots could easily be obtained at the designed displacement, and reports show that on service this is actually exceeded. The design of these vessels was begun late in January, 1915, and the order for one ship (*Courageous*) was placed with Messrs. Armstrong and the other (*Glorious*) with Harland and Wolff, the latter making their own machinery and Messrs. Parsons sup-

plying the machinery for Messrs. Armstrong's ship. It was intended that these vessels should be built in a year, or as near that as possible, but this was not realized, and the ships were both commissioned in October, 1916.

On her commissioning trials, the *Courageous* worked up to full power, and while steaming during the trials at full speed she met very heavy weather. Some signs of weakness were shown at the fore side of the forward turret where there is an inevitable discontinuity of longitudinal strength, and some doubling plates were accordingly added to the *Courageous.* Her sister vessel *Glorious* was in commission for over a year before similar additions were made to her, although no signs of weakness were shown. This incident shows that the very high speed obtained on trial, reaching 32 knots, should hardly be maintained against head seas in heavy weather.

The *Furious* was similar to, but a modification of, the *Courageous* and *Glorious,* having about the same length and the same machinery, but the form of midship section was somewhat different, having a more pronounced bulge and a simpler form of main framing and structure of the hull. The armament also was different, each turret, instead of having two 15-inch guns, was arranged to carry one big gun of 18-inch bore, although arrangements were made to substitute pairs of 15-inch guns, if thought desirable later.

The order for this ship was placed with Messrs. Armstrong about two months after that of *Courageous,* and she was to be finished in the shortest possible time. Early in the spring of 1917, however, the necessity for having fast aeroplane-carriers became very obvious, and it was approved to fit *Furious* for this purpose. This entailed doing away altogether with the fore turret and making other considerable alterations. A large hangar was built on the forecastle deck, and a flying-off platform 160 feet long was arranged on the roof of the hangar, which was designed to house about ten machines. Later it was decided to remove the after turret as well, and a flying-on deck 300 feet long, extending from the funnel aft, was constructed. The secondary armament, which had consisted originally of eleven 5½-inch guns, was retained, with the exception of one gun; the remaining 10 guns being rearranged. Four sets of triple 21-inch torpedo tubes were fitted on the upper deck aft, and one pair each side on the upper deck forward. After these alterations were completed the ship was tried and commissioned in July, 1917, a speed of 31½ knots being obtained with 94,000 shaft horsepower at 330 revolutions.

LIGHT CRUISERS (FIG. 8 AND FIG. 9).

Following upon the previous light cruisers of the "*Town*" classes, the particulars of which have already been published, a very important departure was made in the light-cruiser design in the program 1912-13, when the "*Arethusa*" class was designed by Sir Philip Watts. The importance attached to speed was specially brought out in this design, and it was decided to instal very powerful machinery with a shaft horsepower of 40,000, and this could only be achieved by adopting a type of engines and boilers closely approximating to those hitherto used for destroyer classes.

In conjunction with high speed a good armament was provided, consisting of two 6-inch and six 4-inch guns, though in the original design the armament consisted entirely of 4 inches. The ship's sides up to the level of the upper deck were protected by specially high-tensile plating varying from 2 inches to 1½ inches throughout the machinery spaces, in addition to the 1-inch shell plating. This arrangement of plating also greatly added to the strength and stiffness of the ship regarded as a girder. Further particulars of the class are given in the Tables. The

Arethusa and other light cruisers were in the action off Heligoland on August 28, 1914, and gave an excellent account of themselves.

In the 1913-14 program the "*Calliope*" class, slightly larger vessels than the *Arethusa*, but with the same power, were decided upon. After considerable discussion regarding the merits of mixed or homogeneous armament, it was decided to give these vessels two 6-inch guns, both on the center line placed aft, and eight 4-inch guns. The protection consisted, as in the previous design, of a 2-inch belt over the shell plating, making a total thickness of approximately 3 inches. Most of this class had practically the same machinery as the *Arethusa*, but Parsons geared turbines were installed in two of them, the *Calliope* having four shafts and the *Champion* having only two shafts. This was at the time a very important experiment, the putting of 20,000 H.P. through gearing being a very bold departure from anything which had been hitherto contemplated. The final results obtained with *Champion* were, however, excellent, and she obtained a speed of 29½ with 337 revolutions and about 41,000 shaft horsepower, this speed being slightly in excess of any of the other vessels of the class at corresponding displacement.

For the subsequent classes I would refer to the Tables, which show a gradual growth in size and power of armament; "*Ceres*" class, Fig. 8, finally having a length of 425 feet and a beam of 43 feet 6 inches, and a normal displacement of about 4,200 tons. These vessels carried five 6-inch guns, all on the center line.

The next class were the "*D's*" the general arrangement and protection of which followed that of the "*Ceres*," except that six 6-inch guns were carried on the center line instead of five. The power was only slightly increased in these ships over the previous classes, but the revolutions were reduced to 275, all these later classes having the twin-screw geared arrangement, and although the displacement of the "*D's*" increased to 4,650 tons, the additional length and the reduction of revolutions enabled the speed of close upon 30 knots of the whole class of light cruisers, "*C's*" and "*D's*," to be practically maintained. The "*Arethusas*" and the "*C*" and "*D*" classes all had oil fuel only.

In addition to these light cruisers, which were all to Admiralty design, two vessels—the *Birkenhead* and *Chester*—which were built at Messrs. Cammell Laird's for the Greek Government, were purchased in 1915. These vessels were considerably heavier than the "*C*" class, and more closely resembled the British "*Chatham*" class. They carried an armament of ten 5½-inch guns. The machinery was modified to burn oil only, in the *Chester*, instead of coal and oil as in the *Birkenhead*, and the resulting increase in power to 31,000 gave the former a speed of 26½ knots.

The "Raleigh" Class.—In the summer of 1915 designs were prepared for a considerably heavier class of light cruisers, more especially designed for ocean work in any part of the world. They were to have a speed of 30 knots and a large radius of action. Various armaments were considered, and it was finally decided to adopt an armament of seven 7.5-inch guns with twelve 3-inch (four being on high-angle mountings). Five of the big guns were placed on the center line, as shown in Fig. 9, and the other two were on the broadsides amidships. The bow and stern guns were superposed, thus giving a fire of four guns, both ahead and astern, and six guns on either broadside. The conditions laid down involved a ship of great length, and these vessels consequently have an overall length of 605 feet. These ships were originally designed to burn oil and coal, but this was subsequently altered in three ships of the class to all oil, the original power of 60,000 shaft horsepower on a four-shaft geared turbine arrangement being considerably increased up to about 70,000 shaft horsepower by the change to all-oil firing. These vessels also

different from the light cruisers referred to above in having modified bulges as protection against underwater attack. The form of the section shows how these were arranged. The protective plating was similar to that of the other light cruisers.

MONITORS. (FIGS 10 AND 11.)

The first vessels of this type to be added (or reintroduced) to the British Navy were the three ex-Brazilian river monitors, built by Messrs. Vickers, Limited, and taken over by the British Government in August, 1914, and renamed *Humber, Mersey* and *Severn*. The particulars of these vessels are given in the Tables. It will be seen that the armament consisted of medium caliber guns, viz, 6-inch and 4.7-inch. These vessels, though lightly built, have done very good service in the war, both on the East African and Belgian coasts.

The need for vessels of the monitor type mounting heavy guns soon became apparent, and I received instructions in November, 1914, to prepare designs of monitors of more substantial structure for sea-going service, but of light draught with good protection and carrying some heavy guns, the light draught combining the advantages of being able to go close inshore and of greatly reducing risk of being struck by a torpedo.

The earliest design was that of the 14-inch gun monitors, four in number, which was commenced in 1914. Four twin-mounted 14-inch guns and mountings were available, and with the very simple form of structure adopted, these vessels were designed and built in six months. They were quickly followed by the 12-inch monitors, which were of similar design but carried pairs of 12-inch guns, generally taken from older battleships. These vessels were also built in about six months. They all had a complete bulge of a form which was of simple construction, with an air-space outboard and a waterspace between that and the ship proper. This bulge was carried well fore and aft, and the ships being of rather bluff form, the speed was somewhat below that estimated; the bluff form of the bulge having a bad effect upon the propeller performance. However, no particular point had been made about the speed, and we had rather to take what machinery we could get in the shortest possible time.

Following on the 12-inch monitors, early in June, 1915, two more vessels were ordered, mounting a pair of 15-inch guns. For these ships, internal-combustion engines, which were well under way, but designed for another purpose, were installed. In September, 1915, two improved 15-inch monitors were ordered and named the *Erebus* and *Terror* (Fig. 10). These were of finer form, of more power and greatly increased speed. These vessels also had an improved form of bulge, though it did not differ essentially from the bulges of the earlier monitors. The *Erebus* and *Terror* were designed to have a speed of 12 knots when fully laden, but they actually are capable of about 14 knots.

Following the earlier 15-inch monitors, some much smaller vessels, each carrying a 9.2-inch gun, were designed, and others again which carried 6-inch guns (Fig. 11). I need not refer at any length to these, as the general particulars are given in the table. A good many of both large and small monitors went out to the Dardanelles in the early part of the war and did very good work, and for a long time they seemed to bear a charmed life, as they enjoyed complete immunity from torpedo attack. Whether this was due to their shallow draught or to the enemy submarines thinking it was useless to torpedo the bulges, I do not know. Later, however, the *Erebus* and *Terror* were both torpedoed: the latter received three torpedoes forward, but got to port all right, though severely damaged, two of the torpedoes hitting forward of the bulge; the third, which hit the bulge itself, did very little damage. The former ship was hit full amidships by a distance-controlled boat carrying a very heavy charge, but

the bulge gave her complete protection and she was repaired in less than a fortnight. It is interesting to note in this connection also that some of our old cruisers of the *"Edgar"* class, which had had bulges added to them very early in the war, were torpedoed in the Mediterranean, but the bulge gave complete protection to the ship proper. They were taken to port and repaired. Of the heavier monitors it may be remarked that of all ships carrying heavy guns these vessels were probably more often in action off the Belgian coast and elsewhere than any of our heavy-gun ships, and they no doubt gave the enemy in occupation of that coast a very anxious time.

DESTROYERS AND FLOTILLA LEADERS.

To give a complete account of the development of the design and construction of our destroyers and flotilla leaders during the war would mean a very long paper, and I think, therefore, I must refer only to the tables and plans (Figs. 12, 13 and 14) which I give. These vessels have gradually increased in size and power, and war requirements have continually added to the weights which they have had to carry, including considerably more fuel, heavier armament, both of guns and torpedoes, depth charges, larger bridges, and other additions. In fact, some of the ships which before the war were 900-ton vessels, now carry an additional 100 tons, which, of course, is a very heavy handicap to these vessels when expected to maintain a high speed. The introduction, however, of the geared turbine has added enormously to the efficiency of the machinery and propellers; the lower revolutions which can be introduced with the gearing enabling us to adopt much more suitable propellers, in addition to giving a greater economy of fuel.

During the war nearly 300 torpedo-boat destroyers and flotilla leaders, which are simply a larger form of torpedo-boat destroyer with improved accommodation, have been added to the Fleet, and the whole class of these vessels has been called upon to do continuous work in heavy weather. They have come through the ordeal with extraordinarily few breakdowns of machinery or other parts of the ship, whilst the duties they have been called upon to perform in combating the submarines, convoying, &c., have been most continuous and varied.

Most of the designs have been got out at the Admiralty, but certain other, including some torpedo-boat destroyers built for foreign countries and taken over by Admiralty have been evolved by the builders, Messrs. Thornycroft, Yarrow, White, Fairfield, Brown, and Hawthorn Leslie. Numbers of these vessels also have been built by firms who had never built a warship before, but the work turned out by them has fully met the Admiralty requirements. The flotilla leaders (Fig. 14), with a deep load displacement of about 2,000 tons and an armament of five 4-inch or 4.7-inch guns, and with their very high speed, might well be described as fast scouts or third-class cruisers.

PATROL BOATS. (FIG. 15.)

Patrol boats naturally follow the torpedo-boat destroyers, as they were specially designed to relieve the torpedo-boat destroyers of some of their strenuous duties of patrolling and submarine-hunting and escort work. The conditions were that they had to be as small as possible, consistent with keeping the sea in all weathers. They had to be fast enough to run down submarines, and they had to have shallow draught and all top-hamper kept low to prevent their being seen at a distance. Economy of fuel was also an important feature, and it was desirable to have them built of mild steel rather than high-tensile steel, to avoid drilling and

Fig.12. "M" CLASS DESTROYER.
ADMIRALTY DESIGN (1914)

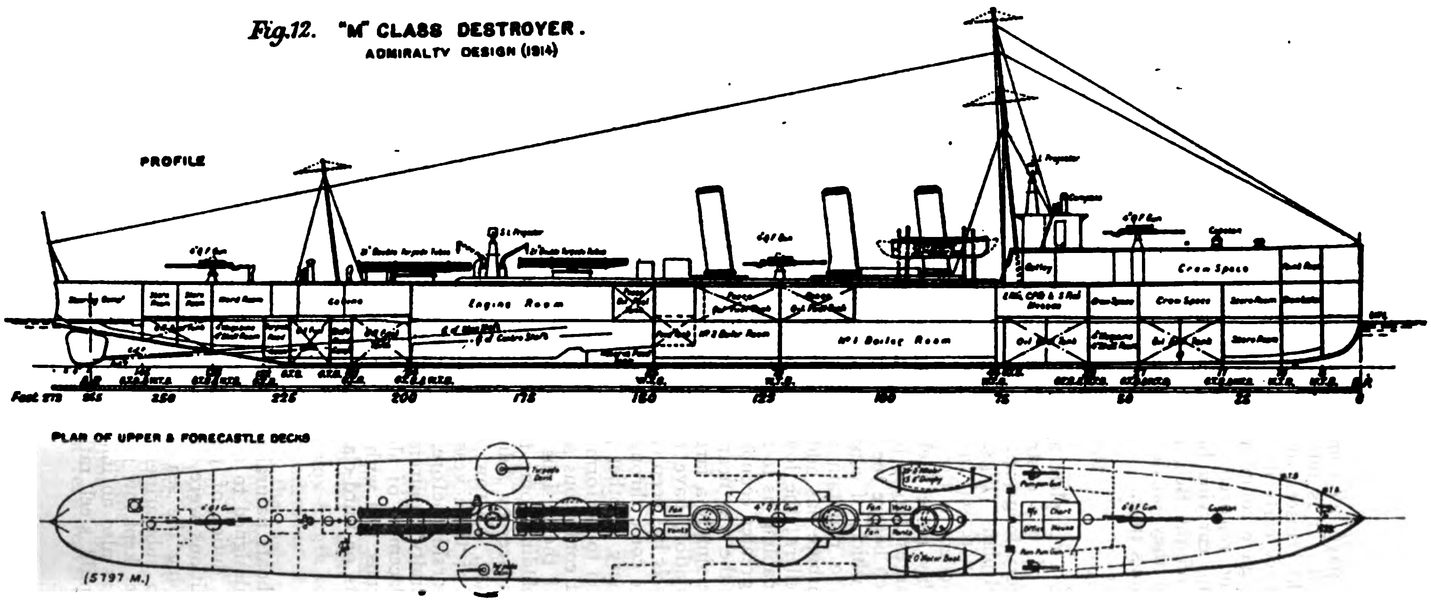


Fig. 13. H. M. S. "VECTIS" ("V" CLASS)

ADMIRALTY DESIGN

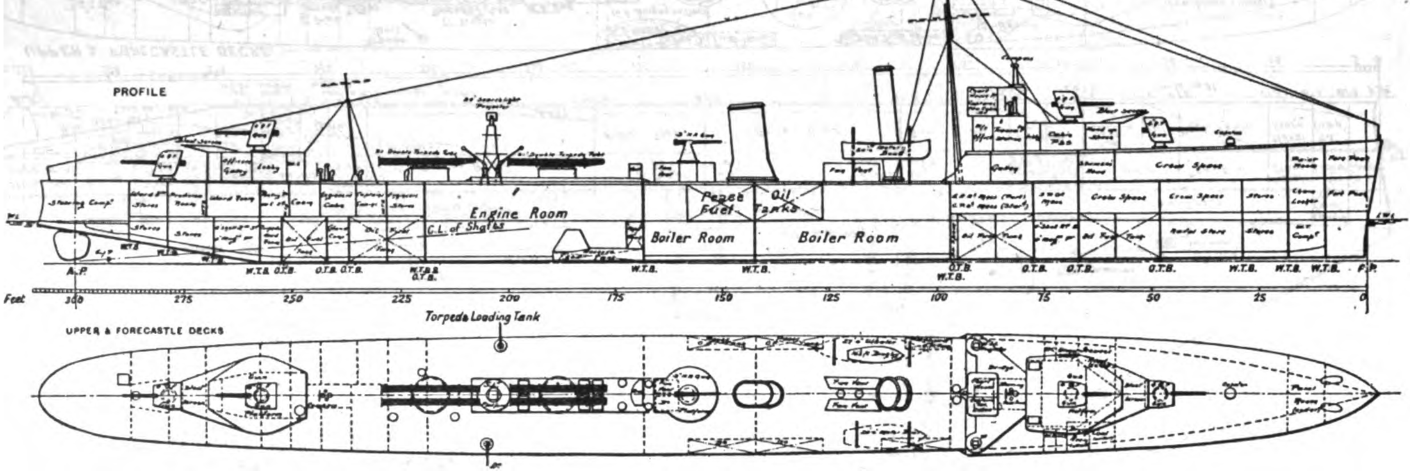
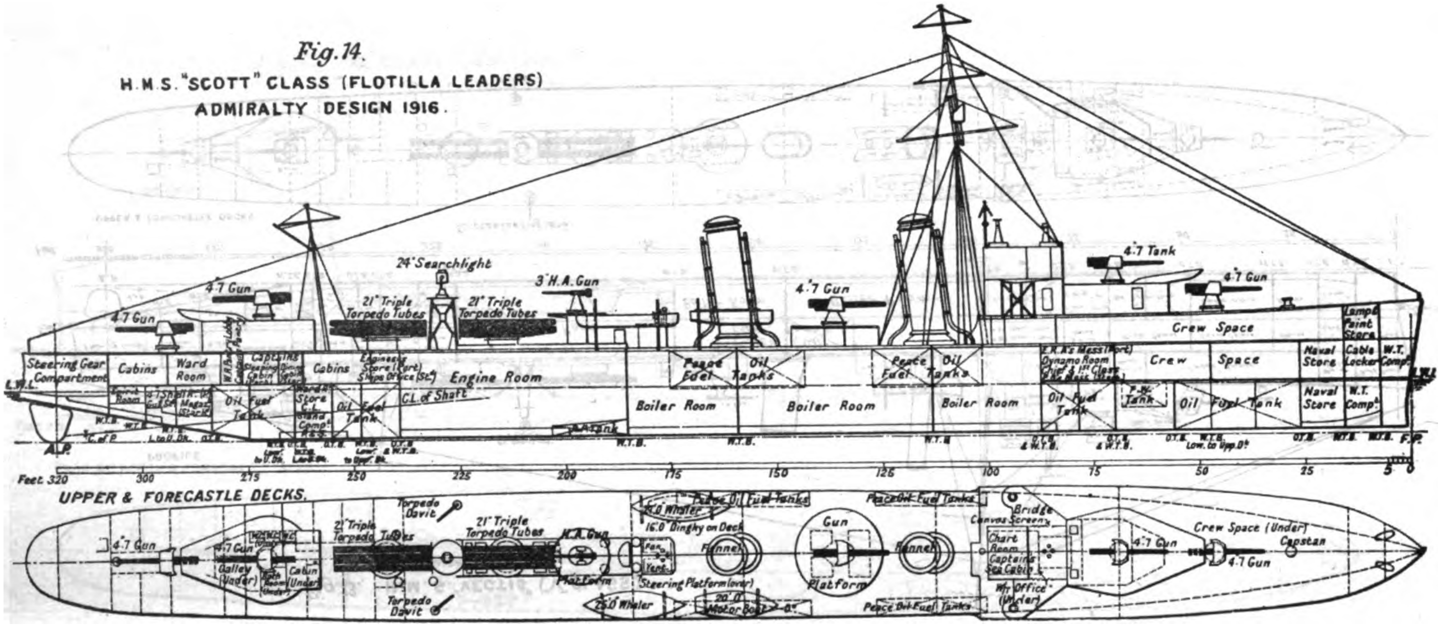


Fig. 14.

H.M.S. "SCOTT" CLASS (FLOTILLA LEADERS)

ADMIRALTY DESIGN 1916.



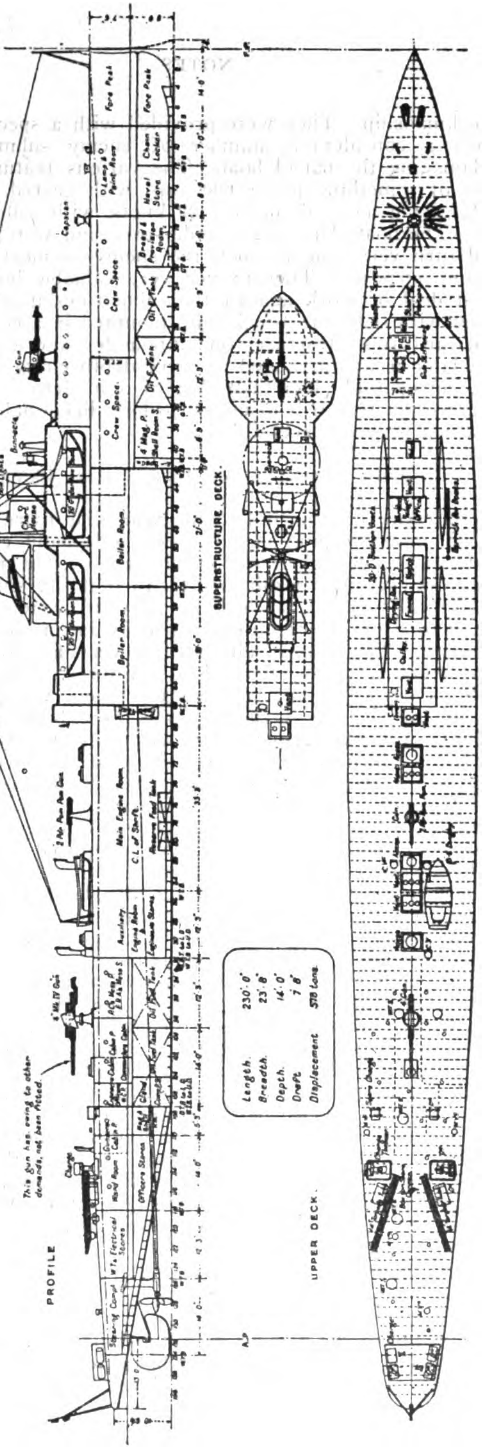


Fig. 15. "P" BOAT, ADMIRALTY DESIGN (AS MODIFIED)
GENERAL ARRANGEMENT

special workmanship. They were provided with a special hard steel ram, with which a considerable number of enemy submarines were sunk without damaging the patrol boat. The various features were combined in a vessel of something under 600 tons, with geared turbine engines of 3,800 H.P., giving a speed of over 22 knots, with 330 revolutions of the propellers. The boats had large rudder area and were cut up aft so that they could turn very quickly upon the enemy—a most important feature for ramming purposes. They proved very valuable boats on service and did a great deal of work against the submarines in all weathers. They were armed with only one 4-inch gun, mounted in a commanding position in the forward superstructure, one 2-pounder and two 14-inch torpedo tubes, and later it was arranged to carry depth charges. Their csot was, of course, considerably less than that of a modern destroyer. Some of these boats were afterwards disguised to look like small mercantile craft—a device which proved successful.

SLOOPS AND MINE-SWEEPERS. (FIGS. 16 TO 18.)

On the outbreak of war it became clear that there would be a great demand for mine-sweeping vessels. A good many coasting and cross-Channel steamers were taken up for this purpose, but more were required, and it was decided in December, 1914, to build 12 single-screw ships of simple design to this end. With the view of hastening construction, it was decided to adopt mercantile practice as far as possible in both hull and machinery. The vessels, although of very fine form, were built of simple construction and under Lloyd's Survey. The boilers were of ordinary Scotch type, and single-screw machinery was provided.

In the end, nearly 100 of these vessels were built, and the armament, which at first was two 12-pounders, was subsequently increased to two 4-inch or two 4.7-inch guns. A great many of these vessels were built in about six months from the order, and the first 36 averaged 24 weeks in building. They proved excellent sea boats, and were used not only for mine sweeping, but also for submarine work, especially for convoying. At later stages some of these vessels were disguised as ordinary merchant ships. They were economical steamers and were able to attain a full speed of 17 knots, with a horsepower of about 1,800 to 2,000 in the earlier, which was increased to 2,500 in the later, vessels. Several of the vessels were mined, and although the damage they sustained was very severe, they kept afloat and were repaired.

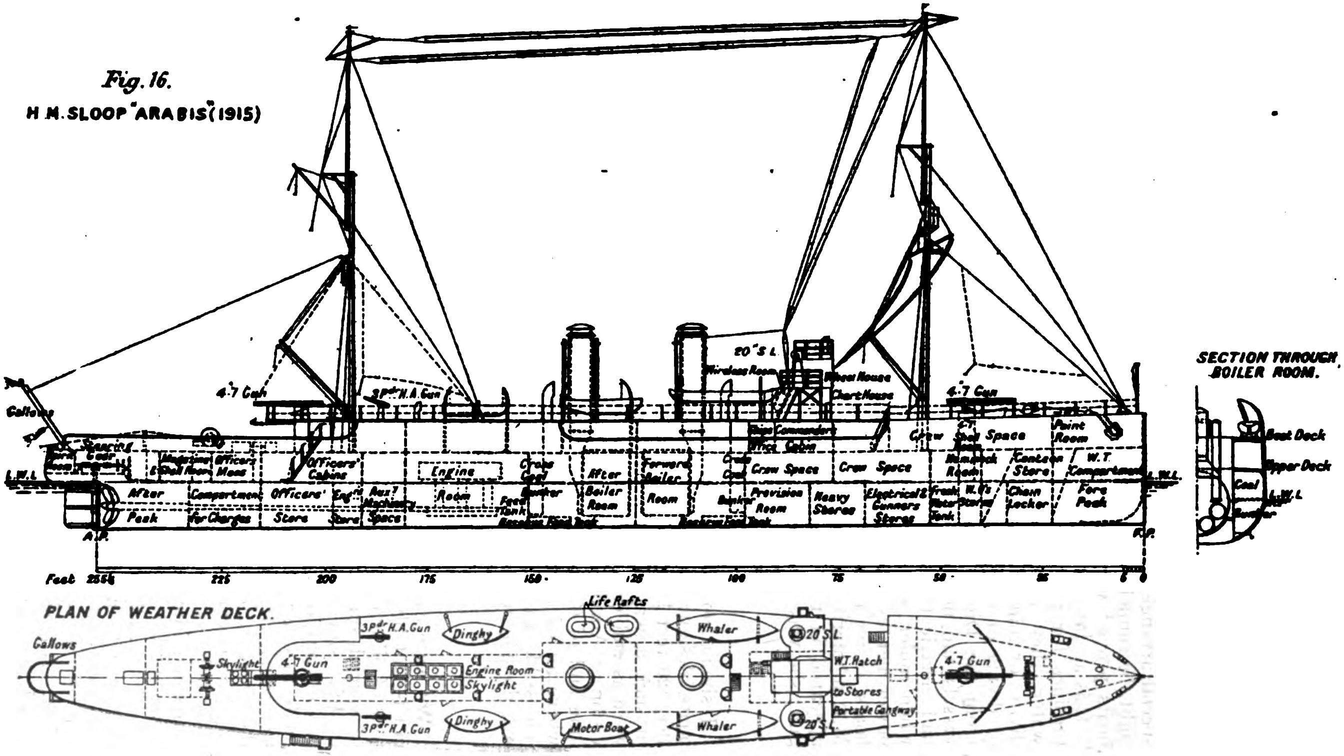
We had the honor of being asked to design and provide some vessels of this type for the French Government, and eight of these were built for that purpose and armed with somewhat heavier armament than our own ships. I understand that the French Government were very satisfied with the vessels. In addition to this, at a later stage, for sweeping in shallow water, some paddle mine sweepers (Fig. 17) were designed at the Admiralty, of which particulars are given in the table. These were 15-knot boats, with draught just under 6 feet 9 inches. They did good work, but were of course not such good sea-boats as the sloops. As there was some danger of mines getting under the paddles, a further design of twin-screw mine sweepers (Fig. 18) was got out, of which particulars are also given in the table. These were vessels of about 800 tons and about 16 knots speed.

SUBMARINES.

As it would make the paper rather too long, I am only giving a table of the particulars of the designs of submarines, but a very large number of these vessels have been added to the Fleet during the war, and some 12

Fig. 16.

H.M. SLOOP "ARABIS" (1915)



different types, some embodying very special requirements and all being improvements on their predecessors. It is interesting to note that we produced the fastest internal-combustion engined submarine in "*J*" class, which attained a speed of over 19 knots. As a still higher speed was wanted for Fleet work, the "*K*" boats were designed with a surface speed of 24 knots. To arrive at this it was necessary to go to steam, and special arrangements of course had to be made for shutting down water-tight the funnels, &c. However, all these difficulties were overcome and the boats proved very successful.

It is an interesting point about these vessels also, that besides the steam turbines for full speed on the surface and the electric drive when under water, they were provided with a Diesel engine for use just before diving or immediately after breaking surface, in order to hasten the time of diving or of getting away quickly after coming up. The transmission from the Diesel engine was through the electric motors, so that these vessels had not only geared turbines for the steam drive, but they also had electric transmission with a Diesel engine and electric battery drive when under water. Although the Germans had the advantage of more power per cylinder in their Diesel engines, we produced faster surface boats, faster underwater boats, and more heavily armed boats than they. M.1 submarine, of which some account and illustrations have already appeared in the Press, was a monitor submarine armed with a 12-inch gun; she was an experimental boat.

"CHINA" GUNBOATS. (FIGS. 19 AND 20.)

I should not omit to mention the two classes of so-called China gun-boats, designed by Messrs. Yarrow. The smaller of these vessels (Fig. 19), 120 feet long and of about 100 tons, were constructed in this country in such a way that the parts could be sent out to Abadan where they were assembled and the vessels re-erected and completed under the super-vision of Admiralty officers. Some of the larger boats (Fig. 20), 230 feet long and of 645 tons, were completed here and went out to Mesopotamia, where all of them were of the greatest service in that campaign. It is impossible to mention all the small craft built, but most useful work was done by motor launches and many other types.

OTHER AUXILIARY CRAFT AND AIRCRAFT CARRIERS.

We were called upon at the Admiralty to design many other auxiliary craft—notably some fast Fleet oilers which were able to carry 5,000 tons of oil and had a speed of 15 knots. We also designed a great number of special smaller craft for all purposes, and we had to take over and modify, sometimes very extensively, a number of merchant ships for various purposes. Probably the most important modifications were those made to vessels taken over and converted into aircraft-carriers, including *Campania, Ark Royal, Engadine, Riviera,* &c. I have already alluded to the *Furious,* but one of the "*Raleigh*" class has also been converted into an aircraft carrier, and is now named *Vindictive.* The *Argus* was originally built as a passenger mail ship of 20 knots and was taken over and converted into an aircraft-carrier with complete flush deck, the funnels being carried aft in long horizontal ducts, discharging the smoke astern. The *Eagle,* alluded to above, is also being converted into a large aircraft-carrier with a somewhat different arrangement; but it would take too long to go into any detailed account of these ships. There is no question that air-craft-carrying will gradually become more important for the Fleet. Altogether during the four years some 2,000,000 tons have been added to the Navy, at a cost probably between 250,000,000*l.* and

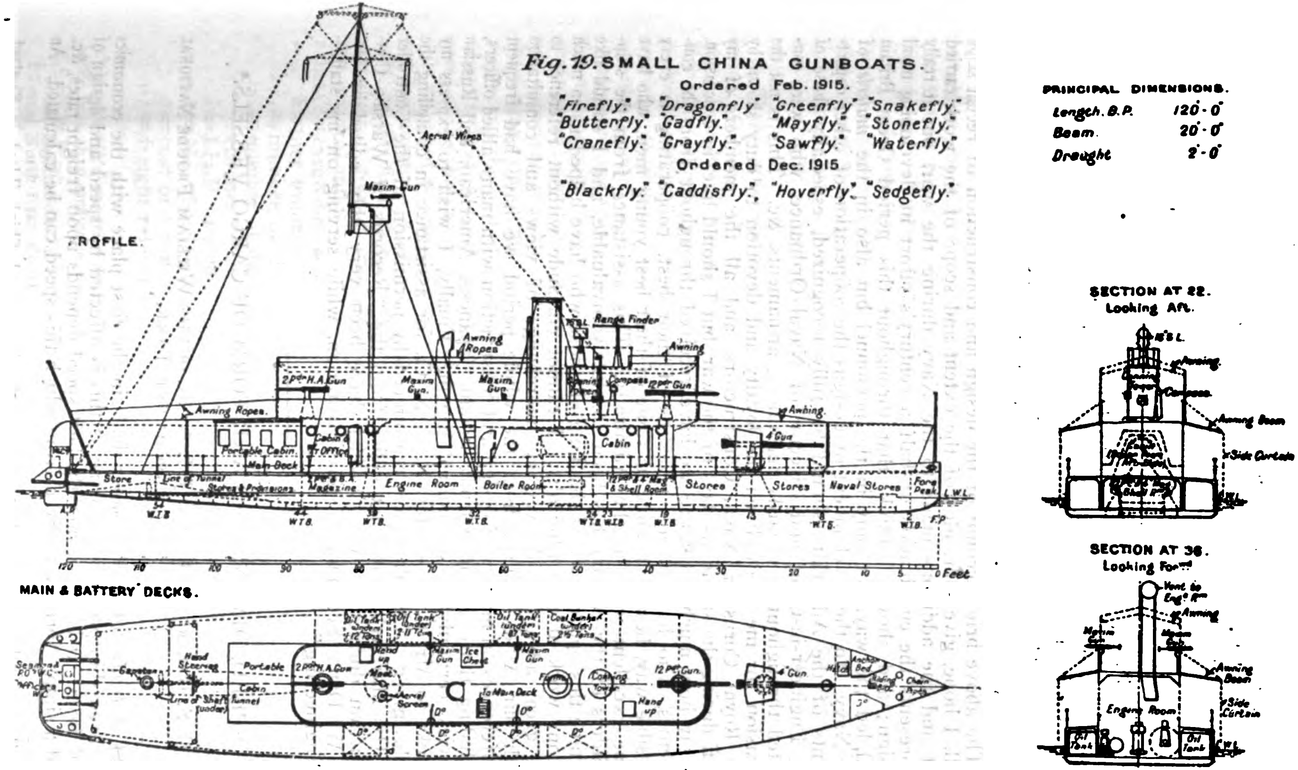


Fig. 19. SMALL CHINA GUNBOATS.

300,000,000*l*. sterling. Reference to the Navy Estimates show that the aggregate sum spent during the four years before the war on new construction amounted to approximately 60,000,000*l*.

AIRSHIPS.

Besides purely ship design and construction, up till the autumn of 1917, as Director of Naval Construction, I was responsible for the rigid airship designs, to which all our existing rigids have been constructed. The responsibility for this branch was in 1917 transferred to a special Airship Production Department, but officers of the Royal Corps of Naval Constructors transferred to the new department continued the work they had been doing on airships.

The above brief account of the design and construction of recent ships will, I hope, give some idea of the extent and scope of the work carried out and the additions made to the Navy during the war, but it really represents the culmination of half a century's effort in developing naval design; and shows, I think, that throughout this period Great Britain had led the way, not only in the personnel but also in the *materiel* of the Navy. In the development of designs the coöperation of other departments of the Admiralty should be fully recognized, especially those of the Engineer-in-Chief and Director of Naval Ordnance, who were consulted throughout as regards machinery, armaments, &c. This is not the occasion for me to dwell on the loyalty and devotion to duty shown by the Royal Corps of Naval Constructors, and all the members of my staff during the strenuous days of the war, but I should like, in conclusion, to say how much assistance we have received throughout from the contractors, who have always given us of their best, coöperating in every possible way, besides lending us some of their best young men, who have done excellent work at the Admiralty. This assistance from the shipbuilders of the country has been absolutely invaluable, and I should like to take this opportunity of thanking all those who have helped us so much in our work at the Admiralty. I cannot conclude without reference to the great advantage derived by interchange of views and consultation with our Allies. Throughout the whole period we have had frequent conferences on matters of design and construction with many allied officers, including those of the French, Italian, Japanese, American, and Russian Navies, certainly to our mutual benefit. Finally, I wish to express my thanks to Mr. C. M. Carter, R. C. N. C., for assistance in compiling the tables given. The plates are reproduced by permission of the Controller of His Majesty's Stationery Office from the "Records of Warship Construction during the War," which have been very ably edited by Mr. R. W. Dana, Secretary of this Institution, while serving on my staff.—"Engineering."

SPEED, DIMENSIONS AND FORM OF CARGO VESSELS.*

By G. S. Baker, O.B.E., and J. L. Kent, of the William Froude National Experiment Tank.

§1. *Introductory.*—The paper deals in the first place with the economics of cargo-ship propulsion, so far as this is affected by speed and design of hull form. It is shown how the best speed depends upon freight rates, &c., and a simple formula is given from which this speed can be calculated. An

* Paper read before the Institution of Engineers and Shipbuilders in Scotland, February 18, 1919.

example in Appendix I gives best speeds varying from 12 knots to 14 knots for a 4,000-mile voyage. The second part of the paper (§10 to §19) gives in detail the various propulsive considerations which have to be borne in mind in settling the area of midship section, longitudinal distribution of displacement, and shape of level lines and body sections. Finally, by permission of the Controller-General of Merchant Shipbuilding, some notes are given on straight-frame ships, based upon test work carried out for His Majesty's Government. Both the latter parts of the paper are illustrated by giving typically good lines and results for curved and straight-frame ships.

§2. *Dimensions and Speed of Cargo Vessels.*—The primary purpose of this paper is to describe and define the necessary features which can or should be embodied in any slow-speed vessel. By slow speed is meant such a speed as is common with ordinary cargo tramps, and may for the purpose of the paper be more precisely limited to speeds not exceeding, say, 14 knots for a 400-foot vessel. As the shape of the vessel necessarily depends on its dimensions and speed, it is desirable first to consider these factors and how they should be determined in practice. The only reason for the existence of the cargo tramp is its earning capacity. The efficiency of the vessel must be measured by the amount of profit earned in a given time with a given amount of capital. For this to be high, the dimensions and speed must be arranged to suit the general service conditions of the vessel.

§3. There are several papers in existence dealing with this phase of design. Of these the authors have examined three:* the first written by Mr. J. Hamilton,† the second by E. Saxton White, B.Sc.,‡ and the third by Mr. John Anderson.§ The fundamental idea in all these papers is undoubtedly correct, but they tackled the problem under very different and restricted conditions, and so arrived at results which are not always in accordance with each other. In all these papers the total yearly expense is calculated for various ships or voyages of certain lengths, and the conditions necessary to reduce this to a minimum (in one case with a standard profit added) are enumerated. But none of these papers takes into account one important item in the ability of the vessel to earn profit, namely, the freight rate. This is fixed by economic factors quite apart from ship design, and the best speed and dimensions are those which make the most of any freight rate. It is a misnomer to call this best speed the economical speed of the ship, as it is not necessarily the most economical result in coal. It is the speed which will give the largest profit per pound capital invested, per day invested.

§4. In the first case, let it be assumed that an existing vessel of displacement $\triangle$ tons is employed on a service of which the average voyage is L nautical miles.

Let C = cargo deadweight.
δ = net tonnage.
n_s = number of days steaming on voyage.
n_1 = number of days in port loading and discharging, repairs, &c., per voyage.
I = I.H.P. of engines for speed V nautical miles per day.
k = No. of tons of coal consumed per day per I.H.P.
q = cost of 1 ton of coal.
f = freight rate of journey.

* Mr. Unwin's paper, read in Newcastle-on-Tyne on January 14, 1919, which is on better lines than previous ones, came to hand too late for any specific reference to be made to it.
† Trans. Institution of Naval Architects, vol. xxiv, page 256.
‡ Trans. North-East Coast Institution of Engineers and Shipbuilders, vol. xxviii, page 29.
§ Trans. Institution of Naval Architects, vol. ix, page 23.

x = tonnage dues in pound per ton for complete journey.
t = sum of brokerage, management, loading, discharge in pounds per ton of cargo, for complete journey.
P = price of vessel in pounds.
yP = sum of insurance, repair, depreciation, wages per year, in pounds.

Then the rate of earning money per day per pound of capital is:

$$M = \frac{(f-t)\,C - x\delta - qkIn_S}{(n_S + n_L)\,P} - \frac{y}{365} \quad . \quad . \quad (1)$$

Writing $(f-t)\,C - x\,\delta = r$

and $R = qklV^3\Delta^{\frac{2}{3}}$ = cost of coal burnt per day in pounds.

I may be written $= lV^3\Delta^{\frac{2}{3}}$

and $n_S = \dfrac{L}{V}$.

Hence $$M = \frac{Vr - qkl\Delta^{\frac{2}{3}}\,LV^3}{(L + n_L V)\,P} = \frac{y}{365} \quad . \quad . \quad . \quad (2)$$

For this to be a maximum, when the speed is changed, the first differential with regard to V must be zero.

Or $$R = \left(\frac{r}{3\,n_S + 2\,n_L}\right) \text{ or } = \frac{(f-t)\,C\,x\,\delta}{3\,n_S + 2\,n_L} \quad . \quad . \quad . \quad (3)$$

If this value of R be substituted in the expression for M, the maximum profit at the best speed in pounds per day per pound invested becomes:

$$M = \frac{2\,R}{P} - \frac{y}{365} \quad . \quad . \quad . \quad . \quad . \quad (4)$$

$$\left.\begin{array}{r}\text{or, the profit per}\\ \text{day earned by}\\ \text{the ship}\end{array}\right\} = \left(\begin{array}{c}\text{twice the cost}\\ \text{of coal burnt}\\ \text{per day}\end{array}\right) - \left(\begin{array}{c}\text{capital}\\ \text{charges}\\ \text{per day}\end{array}\right)$$

§5. Equation (3) shows how the cost of coal burnt per day R should be varied according to the freight rate f, and how the coal burnt per day, and therefore the best speed, should rise and fall with length of journey. Equation (1) shows the importance of reducing n_l the number of days in port to a minimum. These formulæ can be put into various forms, but the above are simplest for handling. In passing, it should be mentioned that Mr. Macfarlane Gray, in the discussion on Mr. Hamilton's paper, gave what amounted to equation (4) above, without the capital charges item.

§6. The second case is the determination of best dimensions. Theoretically, the treatment may be the same as before, *i.e.*, the first differential of M with regard to displacement Δ must be zero. But since almost each term in equation (1) is dependent on displacement, the resulting equation is too cumbersome to be of use. The best procedure, therefore, is to take equation (3) and find the appropriate R and V for several displacements, and then to find which displacement gives the highest value of M in equation (4) for all reasonable freight rates and coal prices.

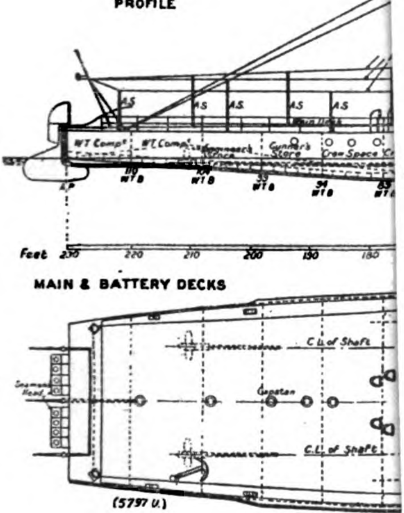
§7. To illustrate this portion of the paper, and to introduce and familiarize the designer with the use of the formula, the detail working of an example is given in Appendix I. Three widely-different freight rates and five widely-different vessels are chosen, and M, the rate of earning profit per pound capital per day, is worked out for all of them. It will be seen that:


Fig. 17. PAD
SECTIO
UPP

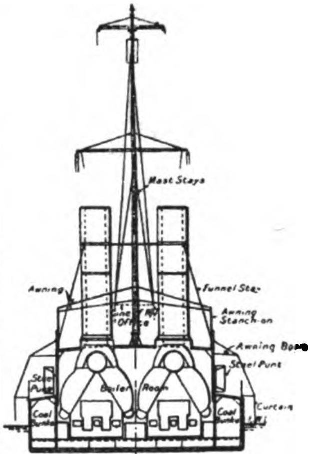
Fig. 20. LARGE CHINA

"Aphis" "Bee" "Cicala" "Cockchafer" "Cricket" "Glowworm" "Gnat" "Ladybird"

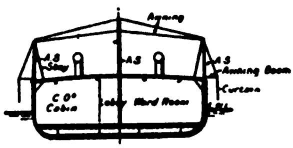
PROFILE



MAIN & BATTERY DECKS



SECTION THRO' BOILER ROOM



SECTION AT 27

(*a*) The profit-earning capacity of both the small vessels is very low under all freight rates, and the smallest vessel can sometimes be worked only at a loss.

(*b*) The profit-earning capacity in the longest vessel is not so good as in the 490-foot vessel for a given price of coal and given freight rate, and the 490-foot vessel is the all-round best one for this voyage.

(*c*) With a 7*s*. 6*d*. freight rate, the speed of this vessel should never exceed 10 knots; with freight at 10*s*. it should remain at 10 knots for coal at 25*s*. per ton, and rise to 13 knots with coal at 12*s*. per ton; and with freight at 20*s*. it should be at least 14 knots, unless coal is more than 26*s*. 10*d*. per ton.

(*d*) The largest vessel is really suffering badly from the long time taken to discharge and load cargo. This time is taken from Mr. Anderson's paper, and it was pointed out in the discussion on the paper that this was high. If n_1 is reduced, R is increased, and the maximum profit-earning will also be increased.

(*e*) It will be understood that for shorter or longer voyages the results may be different. This can be seen by varying n_s in equation (3) for R. A shorter journey increases the best speed for a fixed price of coal, but the amount would depend upon several circumstances.

(*f*) By putting $2\ R = \frac{yP}{365}$ the condition for working at no profit is obtained, R being defined here by formula (3).

§8. The cost factors x, t, y and P are all known with fair accuracy for vessels employed on fairly regular trade, and although q, t and P are liable to vary, they are quite definite, and it is a comparatively easy thing for any designer to find the limiting speed appropriate to average limiting values of freight rate f and coal value q. The vessel should be designed to obtain the highest speed appropriate to these average values. With a good form the propulsive efficiency will then always be good at lower speeds, and good profit returns will result. If the design is arranged for average speed or average freight rate and coal price, the vessel may lose some or all of the advantage of any rise in freight above the average, because of its bad propulsive performance when the speed is increased.

§9. In all calculations of this kind, a number of assumptions are made, and some known factors are neglected, and due weight should be given to these when selecting the speed for any voyage or the displacement and speed for any trade route. For example, it is known that:

(1) The larger a vessel the more seaworthy it can be made, and the less is its speed affected by bad weather.

(2) Reserve coal to meet bad weather must be carried, and the proportion which this reserve bears to the whole decreases both with speed and with displacement.

(3) Relatively greater facilities are required for handling cargo in large vessels than in smaller ones.

(4) Greater coaling facilities are required for large vessels.

It has been assumed that all the vessels on each voyage have full cargoes. Loss of cargo is equivalent to smaller r, and therefore reduces R, *i.e.*, tons of coal burnt per day, which means running at less than full profit. It would be necessary to work this out in detail to tell whether the large or small vessel is most affected, but presumably the larger one is most liable to run under these conditions, as it deals with larger weight of cargo.

(5) The assumption that the indicated horsepower varies, as the cube of the speed does not hold good for many cargo vessels now built, or for any ordinary cargo vessel if it is forced to speeds in the region of 14 knots (on a length of 400 feet). The formula will in this case overrate the proper speed to a certain extent.

§10. *Ship Dimensions.*—It will be seen from the foregoing that actual dimensions do not enter into the problem of the earning capacity of a ship, except as they affect deadweight, displacement and economy of propulsion. Two at least of the principal ship dimensions, namely, length and draught, are so involved with port facilities that in many cases these facilities, and not propulsion at all, define what is possible with both length and draught, and the naval architect's usual problem is to produce the least resistful form of given displacement without altering these dimensions. At this stage it would be useful to state the various forms of resistance which a cargo vessel experiences and their relative magnitude. At quite low speed there is a small amount of wave making, a little eddy formation, and a great deal of skin friction. The latter amounts to about 80 per cent of the whole. At higher speeds, *i.e.,* at what might be called the limiting practical speed,* the skin friction should be about 70 per cent to 75 per cent of the whole, and the wave making most of the remaining 30 per cent to 25 per cent. But in all tramps there is a small amount of eddy formation on the upper levels at the stern near the body post; this point is dealt with later (§15 and §16). The waves created consist of divergent sets at each end, and transverse waves. Economy in wave making can be affected by so arranging the shape that the bow transverse waves tend to cancel the stern transverse waves at such speeds as these may be important.

§11. *Wetted Surface.*—The smaller the wetted surface per ton of displacement is kept, consistent with non-wave-making, the better will be the propulsive performance. Calculations have shown that this is remarkably non-sensitive to such factors as proportion of parallel body in length, and prismatic coefficient, but depends mainly upon the cross dimensions. With fixed beam, the wetted surface per ton will drop steadily as draught is increased, and a similar drop is obtained if draught is fixed and beam increased. The best ratio of beam to draught is that which gives precisely the same reduction of skin per ton for either beam or draught variation. In Appendix II a formula for this ratio has been obtained in the case of a simplified ship form. This formula has been compared with results obtained by calculation for a number of models whose principal dimensions have been widely varied, and found to give results about 6 per cent high, with parallel body from 25 per cent to 50 per cent of the length. The ratio given by the formula must be regarded as a sort of datum. When the $\frac{\text{beam}}{\text{draught}}$ of a ship is materially less than that given by the formula (corrected for the above 6 per cent), there is something to be gained by increasing beam and decreasing draught. When the displacement is constant and the ratio $\frac{\text{beam}}{\text{draught}}$ is near that obtained by the formula, it may be changed to a quite fair extent without any variation of wetted surface. As a matter of interest, this ratio is about 2.8 for some tank forms of 30 per cent parallel body, and somewhat less than this for forms of 50 per cent parallel body. These proportions require to be borne in mind in settling the shape of midship section.

§12. *Midship Section Area and Shape.*—The displacement and length being fixed, the settlement of these items is the first step in fining out the form. For a given speed and displacement there is a midship section area most appropriate for low wave-making resistance. This area A can be found from the simple formula:

$$A = \frac{6.29 \triangle P2}{V2} \text{ square feet.}$$

* i.e., the highest speed possible without any abnormal increase in $\frac{\text{resistance.}}{\text{(velocity)}}$.

Where V is the speed in knots, Δ is the displacement in tons, and P* is a wave-making constant. Most appropriate limits to the values of P* for low wave-making can be obtained from Fig. 1, in which the unshaded portions give minimum resistance.

Putting these values of P in the above equation, the limiting values of A are obtained.† Experiments on models with different midship-section coefficients‡ point to the employment of the largest possible midship section consistent with the other features of the design in vessels of this type, and this conclusion is supported by the general test work of the Froude tank.

With vertical sides from the load water-line to from 4 feet to 5 feet above the keel, associated with a very small rise of floor (or none at all) and a bilge curve of radius of from 4 feet to 5 feet, a coefficient of 0.98 can be obtained.

From an economical propulsion point of view, a rise of floor has the one advantage of permitting easier bow and buttock lines to be worked when there is a considerable amount of parallel middle body, but if a larger radius of bilge and quite flat floors are used, equally good results can be obtained, and it has been found possible, by careful design of the ship at the sections where the entrance and run join the parallel body, to work a square corner to the midship section without any appreciable loss in effective horsepower. Such a shape abolishes the need for a bilge keel, but may present difficulties in constructional design and in practical use. The shape of section being settled, and the draught being already fixed by port facilities, the beam must be arranged to give an area of section A inside the limits already calculated, or as near as possible to these. It should be noted that, as soon as this is fixed, the prismatic coefficient is similarly fixed and the wave-making characteristics are determined. When it is thought desirable, the aim of this section can be achieved by the tentative process of assuming midship area, length, and prismatic coefficient, and seeing what P value is obtained and how this comes on Fig. 1. If it is in a bad place the assumed data should be adjusted.

Longitudinal Distribution of Displacement.—The next step is to decide upon the most economical longitudinal distribution of displacement which, while giving a reasonable position for the longitudinal C.B. will require the minimum horsepower.

§13. *Parallel Middle Body.*—It is generally advantageous, both from the point of view of space and of low cost of construction, to work as much parallel middle body as possible. To determine the greatest length of perfectly parallel body, consistent with low residuary resistance at the required speed, reference can be made to the papers§ giving results of experiments made at the William Froude National Tank. These papers deal with experiments on models of ships of the average dimensions of tramp steamers, and by choosing the set of experiments on models nearest in prismatic value to that already settled (§12) a very fair estimate can be made of the greatest percentage length of parallel middle body to be safely used for the service speed. Similar experiments carried out by Mr. Taylor support these results, and our work of the last few years shows that they can be used with confidence. For vessels of greater or

* The origin and meaning of this constant are given in Trans. Institution of Naval Architects, vol. lv, Part II.

† For a 490-foot vessel of 18,200 tons at 12 knots, or $\frac{V}{\sqrt{L}} = 0.54$ the prismatic coefficient should be above 0.73, to give P a little less than 0.471. That taken for Fig. 2 is 0.77.

‡ Trans. American Society of Naval Architects and Marine Engineers, 1901. D. W. Taylor.

§ Trans. Institution of Naval Architects, vols. lvi and lvii.

Fig. 1. RELATION BETWEEN PRISMATIC COEFFICIENT, LENGTH AND SPEED FOR HIGH & LOW WAVE MAKING RESISTANCE.
The shaded portion indicates high wave making.

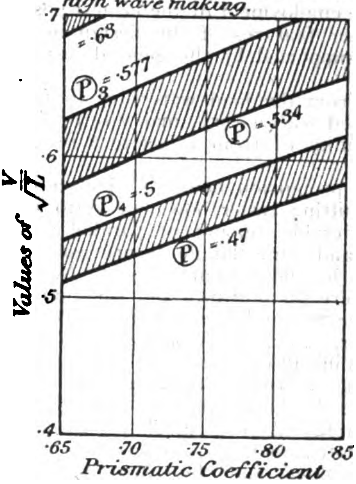
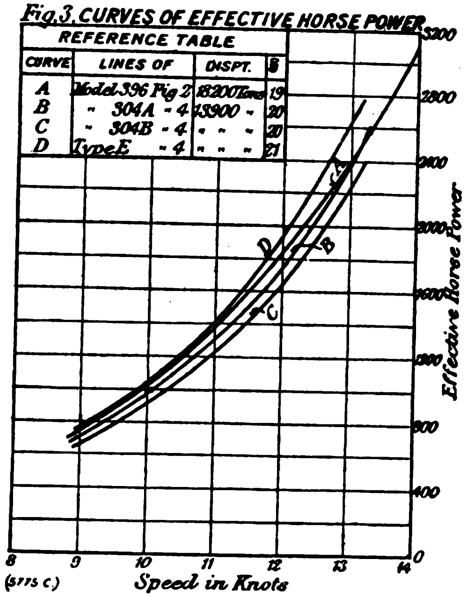


Fig. 3. CURVES OF EFFECTIVE HORSE POWER.



less proportion of beam to length than is usual, the results may be slightly different. The result of an investigation of beam effect will be presented at the spring meetings of the Institution of Naval Architects this year. In passing, attention might be drawn to the very large waste of power that may result from a bad selection of length of parallel for a given speed in short full ships. This is very clearly shown in the classical experiments of Dr. William Froude, Transactions of the Institution of Naval Architects, and the Teddington tank parallel-body series "J" and "K" described in the Transactions of the Institution of Naval Architects, vol. lvii.

§14. *Ratio of Length of Entrance to Length of Run.*—When the percentage length of perfectly parallel body has been settled, the relative amounts of the remaining length of the ship which can be put into entrance and run should be determined. Complete experiments† made at the William Froude National Tank, covering a range of prismatic coefficient of 0.85 to 0.66 with 10 per cent, 30 per cent and 50 per cent of parallel middle body, enable a close estimate of the best ratio to use for the particular speed at which the ship is to run, in order to get minimum horsepower. For high proportions of parallel body (50 per cent) an $\frac{\text{entrance}}{\text{run}}$ of about 0.9 is best, and there is a small advantage in working the entrance shorter than the run even with 30 per cent parallel middle body. The prime cause of this advantage in a low ratio of $\frac{\text{entrance}}{\text{run}}$ is the avoidance of eddy making at the stern by the longer run obtained, and provided this is attained, the resistance is not very sensitive to this ratio, and this permits of a certain amount of freedom in adjusting the longitudinal C.B. Tank experiments have shown that for vessels of ordinary proportions the minimum length of run required is given by $4.1 \sqrt{\text{midship section area}}$ (see also §16).

§15. *Prismatic Coefficient of Entrance and Run and Curve of Areas of Sections.*—The designer is now in a position to draw the curve of areas, which will satisfy the length of entrance, run and parallel body, and the prismatic coefficients and displacement already fixed. He will probably prefer to base the first rough shape of the area curve on that of some particular ship not greatly different in dimensions, whose trial results he may have. Guidance as to the best modifications to the form of this curve can be obtained from the published results of model experiments made at the William Froude National Tank and by Taylor and Sadler, in order to determine the effect of change in prismatic coefficient and shape of the ends. The results of the tank experiments can be found in sets F, G and H, Transactions of the Institution of Naval Architects, vols. lvi and lvii. Generally speaking, (*a*) a somewhat full shoulder to the entrance is not detrimental if the speed is low, but a full shoulder to the run may result in an increase in power, due to either eddy or divergent-wave formation. (*b*) The after end of the run should not be cut away to get a fine finish, but the area curve should be carried out with a fair curve to its end, with no marked change in shape just before the body post. (*c*) Every endeavor should be made to avoid a quick change in the slope of the tangent to the curve, and the maximum slope of this tangent to the base should be kept as small as possible. (*d*) When the prismatic coefficient is below 0.7 and the speed above $P = 0.5$ (*i.e.*, 11.2 knots for a 400-foot ship) the curve of areas of sections of entrance should have a little hollow in it, and this should die out altogether for a prismatic coefficient of 0.8 at all speeds.

† Trans. Institution of Naval Architects, vol. lv.

§16. *Load Water-Line.*—For all tramp steamers this should have a slightly convex or straight entrance. The shoulder of the fore end where it runs into the parallel body can be fairly blunt if necessary. The higher the speed required, the finer must the entrance angle be made, and the greater must be the distance from the stem before it turns into the parallel body. At low speeds, say 9 knots for a 400-foot ship, 60 degrees entrance angle (side to side) can be accepted. By straightening or hollowing the load water-lines to produce a finer end, a lower "bow breaker" and smaller divergent waves are formed. Yet it does not follow that the effective horsepower will be lower if the speed is small in comparison with the ship's length. An increase in power of from 4 per cent to 5 per cent may be the result of working a hollow line forward, although at higher speeds (usually beyond the limits of tramp steamers) the influence of the finer end always shows itself in diminished resistance. At the after end, the angle between any tangent to the load water-line and the middle line of the ship, should not exceed about 20 degrees in order to avoid dragging dead water with the vessel. It is, of course, not always possible to keep the line so fine, and should a greater angle than 20 degrees become a necessity, the curve must be so drawn that the maximum slope is as far aft as possible, consistent with finishing at the rudder post without any marked change of curvature or hollow in the curve. It is not generally realized that at low speeds a fine end aft, obtained by filling out in front of it, is quite as conducive to eddy making as a full finish, as the stream lines tend to break up whenever the rate of expansion exceeds that given by a level boundary of 20 degrees to 25 degrees. It is in this connection that the main propulsive advantage of a cruiser stern is obtained. With ships of either considerable beam or high prismatic coefficient, a well immersed cruiser stern enables a reasonable slope to be given the load water-line and the lines immediately below it, and thus eliminates eddy formation in this area, with a consequent reduction of about 3 per cent of the total resistance. The more the feature is immersed, the better is the result obtained.

§17. *Cross-Sections.*—Experiments have shown that there is some advantage in departing from the usual U-sections forward, and adopting sides sloping inward towards the middle line from the load water-line, with well-rounded heels. The displacement lost at the heel of the sections is regained by working the full water-line suggested. This allows the water which is endeavoring to find its way under the form along bow or diagonal lines, to do so with a minimum rate of change of curvature of flow. To attain this same end it is good to commence easing the flat floor of the parallel midship body into the uprising entrance and run sections as soon as the limit of the parallel body has been reached. The area of section can quite easily be maintained by keeping the full beam at load water-line and the levels immediately below it for some distance forward of the parallel-body limit. This same type of section is required where the parallel body merges into the run, and further aft the sections should be of V form, in accordance with Mr. R. E. Froude's dictum. Double curvature should be kept to a minimum. Club-footing the aftermost sections has not been found to exercise any noticeably bad effect on the horsepower.

§18. *Level Lines and Contours.*—There should be a fairly uniform change in the slope of the level lines from load water-line to keel. It is known that, at the fore end, the water flows approximately along bow lines near the keel. These lines will, therefore, give the best indication of the actual stream lines in this neighborhood, and they should be kept as easy as possible for some little distance transversely on each side of the keel. This can be done either by hollowing the lowest levels at their forward extremities, or, preferably, by cutting away the forefoot of the

bow contour. This tends to diminish the "bow breaker," and also decreases the wetted area, and consequently the skin-friction resistance, while sacrificing practically nothing in the way of useful space. The shape of the upper portion of the bow contour has very little effect until wave-making speeds are reached, when forward rake gives slightly longer and easier level and bow lines. At the after end, some hollow is required in the lowest levels to obtain V-shaped transverse sections, but with the early

EXPENSES OF RUNNING VESSELS OF VARIOUS SIZES.

Ship length, L in feet	250			330			410			490			570		
Displacement, Δ in tons	3,030			6,525			11,520			18,360			27,310		
Cargo deadweight, C in tons ..	1,390			3,890			7,360			12,000			18,360		
Net tonnage, δ in tons	700			1,660			3,210			5,400			9,780		
Since $t = 0 \cdot 175 (f - t)$ becomes 0·825, 0·325, and 0·2 lb. respectively for the three different freight rates, and															
$(f - t)$ C with f = 20/– =	1,147			3,110			6,070			9,900			15,160		
with f = 10/–	452			1,265			2,390			3,900			5,980		
with f = 7/6	278			779			1,470			2,400			3,685		
$x\delta$ where x = 1/3 per ton	44			137			200			340			610		
$(f - t)(C - x\delta = r)$ with f = 20/– =	1,103			2,973			5,870			9,560			14,550		
with f = 10/–	408			1,128			2,190			3,560			5,370		
with f = 7/6	234			642			1,270			2,060			3,075		
Number of days to unload and load, including percentage for repairs for each voyage . = n_L	2·5			4·25			6·8			9·0			13		
With V in knots .. . =	8	12	10	10	12	14	10	12	14	10	12	14	10	12	14
n_s .. . . =	20·8	13·9	16·6	16·6	13·9	11·9	16·6	13·9	11·9	16·6	13·9	11·9	16·6	13·9	11·9
$3 n_s + 2 n_L$ . .. .. =	67·4	46·7	54·8	58·3	50·2	44·2	63·4	55·3	49·3	67·8	59·7	53·7	76·0	67·7	61·7
$\frac{r}{3 n_s + 2 n_L}$ = R = f = 20/– .	16·4	23·7	20·1	51·0	59·2	67·2	92·6	106·0	118·8	141·0	160·2	178·0	191·6	215·0	236·0
f = 10/– ..	6·05	8·74	7·45	19·35	22·5	25·4	34·6	39·6	44·4	52·5	59·6	66·3	70·7	79·4	87·0
f = 7/6 ..	3·47	5·0	4·27	11·0	12·8	14·5	20·0	22·9	25·7	30·4	34·5	38·3	40·5	45·4	49·8
Estimated coal burnt per day, in tons .. .. .. =	15·5	33·4	19·3	28·1	48·5	77·0	37·5	64·8	102·6	48·7	84·0	133·0	62·5	108·0	171·0
Corresponding q or cost of coal in shillings and pence per ton f = 20/– ..	21/2	14/2¼	20/1	36/4	24/5	17/5	49/4	32/8	23/2	57/10	38/2	26/10	61/3	39/10	26/7
f = 10/– ..	7/9¼	5/3	7/9	13/10	9/3¼	6/7¼	18/4	12/3	8/8	21/6	14/2	9/11¼	22/7¼	14/8¼	10/2
f = 7/6 ..	4/6	3/–	4/5¼	7/10	5/3½	3/9¼	10/8	7/1	2/6¼	12/5¼	8/3	5/9¼	12/11	8/5	5/10
Capital cost in pounds (P) . =	26,700			44,700			66,800			104,900			153,700		
Profit per pound per day, or $M \times 1000 = \left(\frac{2\pi}{P} \frac{y}{365}\right)$ f = 20/– .	0·85	1·4	1·13	1·9	2·26	2·62	2·39	2·79	3·17	2·31	2·68	3·02	2·11	2·42	2·89
f = 10/– ..	0·076	0·227	0·181	0·489	0·63	0·76	0·658	0·807	0·951	0·623	0·763	0·889	0·545	0·655	0·757
f = 7/6 ..	–0·12	0·0	0·06	0·115	0·195	0·27	0·22	0·31	0·39	0·20	0·28	0·35	0·15	0·215	0·27

upward movement of the floors mentioned in §17 this hollow can be kept within limits, and in twin-screw ships with the deadwood cut away at the post, can be reduced to nil. The thing to avoid is a marked convex curve at the shoulder, followed by a steep slope as the level line approaches the

middle line, as this gives very steep buttocks. Above the propeller aperture full advantage should be taken of the extra length of the level which now run to the rudder post. Filling out these levels prevents any eddy formation in this area, and effects an appreciable reduction of resistance in the same way as does a cruiser stern.

§19. To illustrate this portion of the paper a design for a 490-foot vessel of 18,200 tons (see Appendix I) has been drawn out, and is given in Fig. 2 This vessel is required to steam at a maximum speed of 14 knots when the freight rate is high, and at 10 knots when freight is low and coal high, Appendix I, &c. The prismatic coefficient taken is 0.77. This evades wave-making up to about 12.3 knots P = 0.471, and gives a midship area of 1,670 square feet, corresponding to a beam of 62.75 feet and draught of 26.83 feet, the rise of floor being nil. Of the total length 30 per cent has been made perfectly parallel middle body. The ratio of length of entrance to run has been taken = 0.75, this gives a length of run = 196 feet, and entrance 147 feet. The angle of entrance of the load water-line has been fixed at 55 degrees, and all the bow sections cut away at the heel, the curve of areas of sections having only a very slight hollow in it. This cutting away has involved the contour of the stem being raised to avoid hollow lines. At the stern the load water-line is 29.5 degrees. The sections have been lifted at the heels as soon as possible aft of the parallel body, and the form has been filled out above the propeller aperture. The curve of effective horsepower for this form is given in Fig. 3. It should be noted that this design is given to illustrate general principles, and that careful consideration is required before it is adopted for conditions other than those assumed. For example, its propulsive efficiency is falling off at 14 knots and rapidly diminishes at higher speeds.

§20. *Straight-Frame Ships.*—This type of vessel only differs from that considered in the preceding paragraphs in having the sections built up of straight lines instead of curves. The same considerations apply with little qualification to this type as to the other. One additional matter, however, required attention, namely, the run and number of the chine or knuckle lines. A variety of forms of this type have been tested at the tank to the order of the Deputy-Controller of Auxiliary Shipbuilding, and the Controller-General of Merchant Shipbuilding, and the authors are indebted to Lord Pirrie for permission to publish the following data derived mainly from such experimental work. The run of the chine, particularly in elevation, is a most important factor. This can be so adjusted that the resistance for the form is within a few per cent of that of the curved section ship. With a single chine form, *i.e.*, one in which the side frames and floors meet at a sharp angle with no attempt to bevel or round the joint, this excess resistance can be reduced to about 6 per cent, or with allowance for bilge keels, which are not necessary on the single chine form, to 3 per cent or 4 per cent. With the chine bevelled away throughout the form, giving a double-chine line, a slightly better result can be obtained. This excess resistance is due to a small amount of eddy formation at the chine. Several single-chine forms have been tested with the sharp angle of the sections replaced by curves of 4-foot radius throughout, and these gave a performance of the same as for the ordinary curved-section type. The lines for a vessel of this type are given in Fig. 4 and the results in Fig. 3. A form similar to this, with sides carried down to the keel line, and having a sharp angle at the joint, gave results some 60 per cent worse. Lifting this knuckle to an increasing amount towards the ends steadily improved results, until a line, as given in Fig. 4, was reached. It was also found that, as for the curved-frame ship, that the rise of floor should increase as soon as possible in entrance and run, but the sides may be kept to full or water-line beam for some little distance into entrance and run.


L
M
N
O
P
Q
R

§21. Similar tests have been made with models in the American tanks, and a number of results have been published. In Fig. 3 is reproduced the effective horsepower curve for the Michigan tank, type E,* a model of approximately the same dimensions and displacement as Model 304. The lines of this model are given in Fig. 4. A form of about the same relative dimensions, but of a different type (developed in the Washington tank) having Y after sections and marked U sections with vertical sides forward, gave results very similar in power to the Michigan model, being slightly worse at 8 knots and 2 per cent or 3 per cent better at 12 knots.

APPENDIX I.

To find the best speed and displacement for a voyage of 4,000 miles, when the freight rate is 20*s.*, 10*s.*, and 7*s.* 6*d.* per ton, capital charges as insurance, depreciation, repairs, &c., per year being one-seventh initial cost ($y = 1/7$), loading, discharge, brokerage and management charges 3*s.* 6*d.* per ton ($t = 0.175$ pound), tonnage dues 1*s.* 3*d.* per ton. Delays for coaling, repairs, holidays, &c., and coal consumed taken as in Mr. Anderson's paper.

For the purpose of the example, the calculation has been made for the five vessels, as used in Table I of Mr. Anderson's paper, and as a result the table is much larger than would ordinarily be required.

Cargo deadweight C has been taken constant for all speeds, but actually would be a little less at high speeds than at low. This could easily be set right in an actual ship, but it makes little difference in comparing different ships, and if necessary could be corrected by slight adjustment of the M value in the last line.

It should be noticed that even fair changes in such items as time for loading and repairs, &c., will not make a serious change in comparative results, *i.e.*, they only want fixing approximately.

For further remarks on the table see §8, and for definition of symbols see §4.

APPENDIX II.

Best ratio of beam to draught for minimum wetted surface per ton displacement, for solid of geometrical form.

The solid is shown in plan in the figure, and is wall-sided of draught "a" feet.

The displacement $\Delta = 2\,a\,b\,(l + c)$ the wetted surface "S" $= 2\,(a + b)\,l + 2\,b\,c + 4\,a\,\sqrt{b^2 + c^2}$. The best dimensions are those for which $\dfrac{d\,\frac{S}{\Delta}}{d\,\Delta}$ is the same for change of Δ either by beam or draught variation.

---

* This is the best form, for which complete data were published by Mr. Saunders in August, 1917. Professor Sadler has published (in November, 1918) some results for a form Y i c, which appears to be better in performance, but the data are not sufficiently complete to enable a definite statement to be made.

$$\text{For a change } \frac{d\,\frac{S}{\Delta}}{d\,\Delta} = -\frac{1}{2\,a^2 b\,(l+c)},$$

$$\text{for } b \text{ change } \frac{d\,\frac{S}{\Delta}}{d\,\Delta} = \frac{\frac{4\,b^2}{\sqrt{b^2+c^2}} - 4\sqrt{b^2+c^2} - 2l}{4\,a\,b^2\,(l+c)^2}$$

and for these to be equal

$$\frac{b}{a} = \frac{2\sqrt{b^2+c^2} + l - \frac{2b^2}{\sqrt{b^2+c^2}}}{(l+c)} = \frac{2c^2 + l\sqrt{b^2+c^2}}{(l+c)\sqrt{b^2+c^2}}$$

For a box, *i. e.*, blunt ends, $\frac{\text{full beam}}{\text{draught}} = 2{\cdot}0.$

For a form with no parallel body $\frac{\text{full beam}}{\text{draught}} = 4$

$$\left\{1 - \tfrac{1}{3}\left(\frac{\text{beam}}{\text{length}}\right)^2\right\}$$

TERRY REDUCTION GEARS.

Terry reduction gears manufactured by The Terry Steam Turbine Company of Hartford, Conn., are again on the market, not having been obtainable during the last year, due to the concentration of this company almost entirely on turbines for the Destroyers. Although made primarily for sale with *Terry* turbines, the gears alone are available whenever a high-grade reduction gear is desired.

In the design of *Terry* gears the same qualities which made the *Terry* turbine so popular, namely, simplicity, reliability and interchangeability, have been incorporated. There are a number of features of design which are particularly interesting.

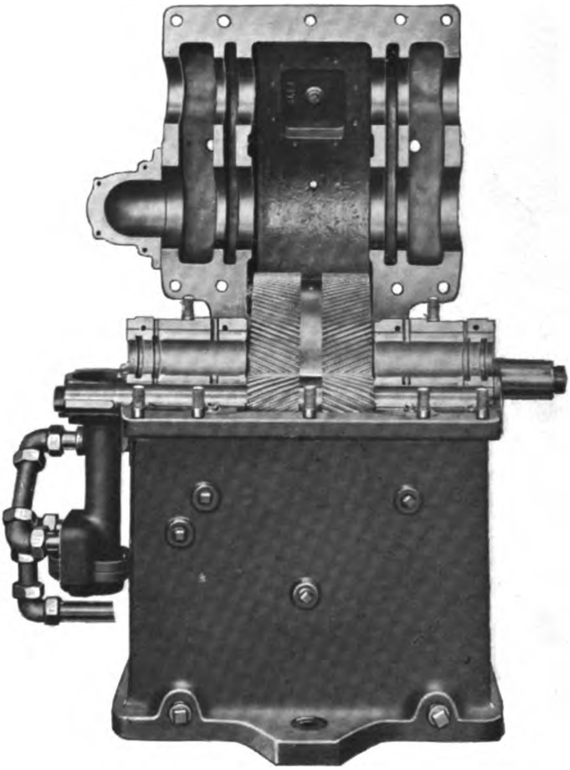
The *Terry* gears and pinion are of the stub-tooth, double-helical type, generated to true form on a Fellows Gear Shaper. The accuracy of this method of tooth generation is such that no grinding or scraping process is necessary to insure perfect contact of tooth surfaces. The gears and pinions are interchangeable, an advantage not often found where the final finish of the teeth is by grinding or scraping.

A well ribbed, double-walled, box-like structure, extending the full depth of the case, forms a rigid support for each pair of bearings. The space between the walls acts as a water jacket for cooling the oil. The ribs between the walls act both as stiffening members and water baffles. The central part of the case, directly under the gears, forms an oil reservoir which contains sufficient oil to supply not only the gears but also the turbine.

The case being of the double-walled, rigid, box-like structure, accurate maintained alignment of the gears is insured. The casing is bored with the aid of fixtures, the dimensions and alignment of the boring being checked by accurate gages. This results in a tight and rigid fit of the bearings.

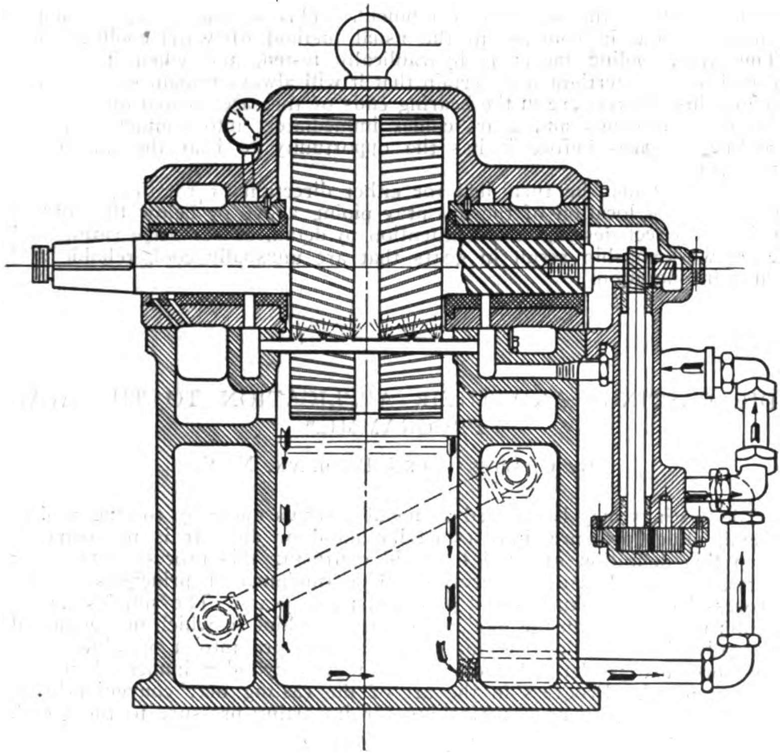
The bearings are of ample size, insuring safe pressures and rubbing speeds. They are split horizontally to permit their replacement without removing the couplings. The bearing shell is of cast iron and, as can be

TERRY GEAR CASE.



Terry Gears, Cover Lifted.

seen in the cross section, is unusually heavy. This is to prevent warping or "pinching-in" from heating, which so often occurs in thinner shells of bronze or similar metal. The lining used is of highest-grade tin base babbitt on both gear and pinion. The bearings are accurately machined to close limits and require very little scraping. Their interchangeability makes it possible, in case of excessive wear, to easily and quickly replace them and restore the original alignment.



CROSS SECTION TERRY GEARS

The oiling system employed is particularly interesting. It is a forced-feed system, the ring-oiling system having been found unsatisfactory for turbine reduction-gear bearings. By reference to the sectional view it will be noted that the oil pump is located well below the oil level in the reservoir, to avoid suction lift, thereby preventing ruin of the gears due to possible failure of the pump to function. The oil is pumped from the reservoir through short, direct, brass piping to a self-cleaning strainer, thence through distributing passages to large, annular oil pockets around each bearing shell, and through the spray pipe from which the oil is sprayed, for lubrication of the gear teeth. The oil-pressure gage is located in one of the above-mentioned annular oil pockets at the most distant point from the oil pump.

The pump and its bevel-gear drive make a complete unit without stuffing boxes or exposed running parts. The unit is so constructed as to be easily accessible. The pump gears may be removed for inspection without disturbing the driving mechanism or oil piping. The bevel gears may also be inspected by removing a small cover. The effectiveness of this lubricating system has been proved by test for, without any water cooling, the temperature of the oil after long runs is less than that found in most gears when running with water cooling.

In spite of the cool operation of the gears, a water-cooling system forms part of the standard equipment. This system is of particular interest, being in contrast to the usual method of water-cooling coils. The water-cooling jacket is hydraulically tested, and when it is once found to be watertight it is certain that it will always remain so. Because the cooling jackets are in the bearing ends of the case, heated oil draining from the bearings and gears comes immediately into contact with the cooling surfaces before it has the opportunity to heat the oil in the reservoir.

The gears may be furnished for either direction of rotation, the only change being location of the oil-spray piping above or below the contact point. Correct design, careful attention to detail, conservative rating and good workmanship result in gears that are unusually cool, reliable and quiet in operation.

DIE CASTINGS AND THEIR APPLICATION TO THE WAR PROGRAMME.*

By Charles Pack,† Brooklyn, N. Y.

Die castings may be defined as metal castings made by forcing molten metal, under pressure, into a metallic mold or die. It is necessary to keep this definition in mind to avoid confusing this process with other permanent mold-casting processes. The fundamental principles of the process have been known and practiced many years. The simplest application is embodied in the modern linotype machine in which molten metal (usually tin-lead alloy) is forced under pressure into a metallic mold. The pressure is derived from a piston and cylinder immersed in the molten metal. Progress in the art of die casting may conveniently be divided into three groups: Machine for imparting pressure to the metal, material for the die or mold, casting alloys.

Casting Machines.—The problem of delivering molten metal under pressure into a die is comparatively simple, when dealing with low fusing-point alloys, as the alloys of lead and tin, but it is much more complicated when dealing with metals of higher fusing points, such as the alloys of zinc, aluminum and copper. Although the art of die casting is comparatively new and, to a large extent, unknown, the records of the Patent Office are replete with patents on the subject.

Fig. 1 shows the Underwood machine patented in 1902; this is probably one of the first machines designed for the production of commercial die castings. The relation of this machine to the linotype-casting machine is clearly apparent. A cylinder and piston are immersed in the molten metal, the application of power to this piston forcing the molten metal, under pressure, into the mold or die. The Doehler machine, Fig. 2,

* Paper read before the American Institute of Mining Engineers, February, 1919.
† Chief Chemist, Doehler Die Casting Company.

Fig. 1. UNDERWOOD DIE-CASTING MACHINE

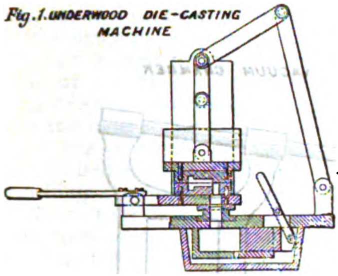


Fig. 2. EARLY TYPE OF DOEHLER MACHINE

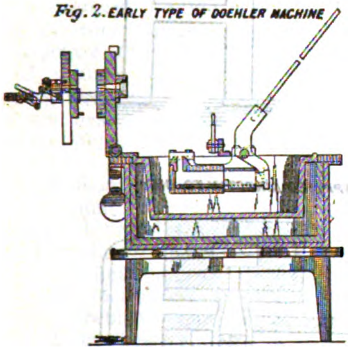


Fig. 3. DOEHLER COMPRESSED-AIR MACHINE.

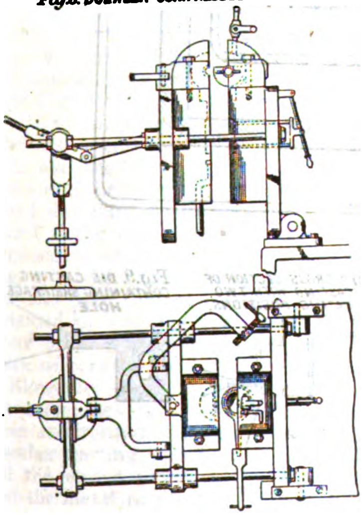


Fig. 4. ANOTHER TYPE OF COMPRESSED-AIR MACHINE.

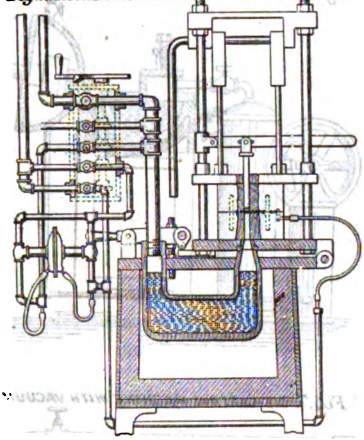


Fig. 5. CHANDLER DIE-CASTING MACHINE

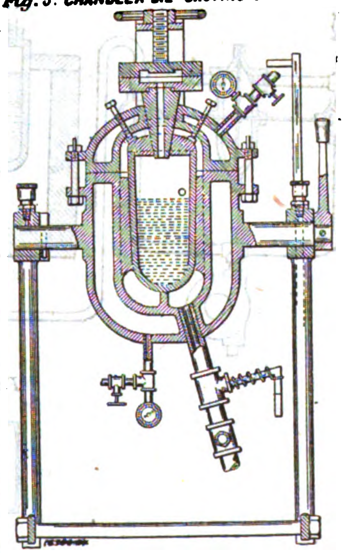


Fig. 6. MACHINE WITH DIE ENCLOSED IN VACUUM CHAMBER

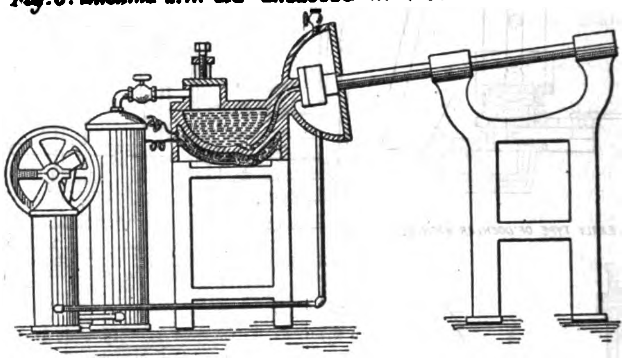


Fig. 7. DIE-CASTING MACHINE WITH VACUUM APPLIED DIRECTLY TO DIE.

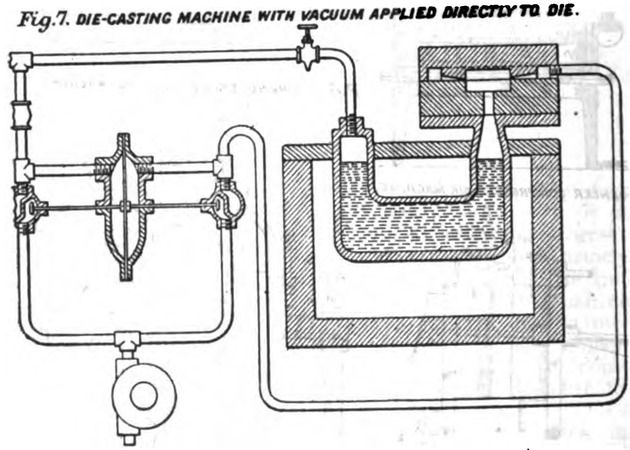


Fig. 8. CROSS-SECTION OF CASTING WITH TWO GATING POSITIONS.

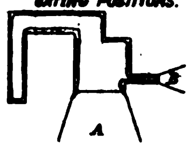
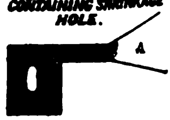


Fig. 9. DIE CASTING CONTAINING SHRINKAGE HOLE.



patented in 1907, is based on the same general principles. This machine is used to a large extent at the present time, throughout the United States, for the production of zinc, tin and lead alloy die castings.

In the machine shown in Fig. 3, patented by Doehler in 1910, compressed air is used for forcing the metal into the die. In Fig. 4 is shown another of this type of machine. Here compressed air is applied to the surface of the molten metal to force it into the die.

In a machine patented by Chandler in 1914, shown in Fig. 5, the principle of the internal-combustion engine is applied for exerting pressure on the molten metal. A charge of gasoline vapor and air is injected into the melting chamber, the explosion of which forces the metal into the die. The writer has never heard of this machine being used on a commercial basis, but it is mentioned to show the various means suggested for forcing molten metal into a die.

Methods Used to avoid Blowholes.—The fact that die castings are made under pressure would suggest, on first thought, dense and homogeneous castings; this impression, however, is not in accord with actual practice. On fracture, the pressure die casting will be found to consist of a dense closely-grained outer stratum and a porous inner stratum. Blowholes of varying size may be expected in the center of the die casting, particularly through heavy sections. Many machines have been designed with the primary object of overcoming this difficulty and producing solid die castings.

Fig. 6 shows an air-operated die-casting machine with the die enclosed in a vacuum chamber. The inventor evidently assumed that the only cause for blowholes in the casting was the presence of air in the die. In Fig. 7 is shown another die-casting machine in which the vacuum principle is applied; here the vacuum is applied directly to the die.

The production of die castings free from blowholes has been the most serious problem controlling die-casting manufacturers. At various times it has been stated that processes capable of producing solid and homogeneous die castings have been developed. If all blowholes in die castings were caused by air coming in contact with metal, the vacuum process would deserve consideration. That the presence of blowholes in some die castings are due to other and more serious causes, the writer will endeavor to prove.

In Fig. 8 is shown a cross-section of a casting that can be gated at A or B; in the best foundry practice the gate A would probably be used. The first metal that goes into the die will chill around the inner walls and take the form shown in the shaded portion. The gate may then become chilled before the inner portion has been filled; this will cause blowholes that no vacuum will eliminate. A similar effect will be produced if the metal was too cold at the time of casting. The writer has produced castings having only an outer shell, similar to that shown in Fig. 8, by limiting the amount of metal injected into the die to a quantity less than that required to make the casting. A similar result may be obtained by running the metal so cold that it will chill the thinner sections of the casting before the heavier sections are completely filled. Lack of pressure will produce the same result.

Blowholes in die castings may also be caused by the phenomenon that we sometimes call "piping." Makers of rolling-mill ingots have often been confronted with this problem. In Fig. 9 is shown cross-section of another casting gated at A. The metal flowing into the die at A will fill the entire mold cavity, assuming all casting conditions to be ideal, but the metal in the thin section adjoining A will chill before the heavier section, so that, the chilling being from the outside, a shrinkage hole will be left in the center. Here, again, no advantage can be gained by the use of the vacuum system.

Dies.—In the manufacture of die castings from zinc, tin and lead alloys, dies made from low-carbon machine steel last almost indefinitely and answer every purpose. In the first attempts to die-cast aluminum, the problem of obtaining a suitable die material presented serious difficulties, which were described by the writer in a paper read before the American Institute of Metals in 1915. This problem, however, has been solved by the use of various alloy steels making possible the commercial die casting of aluminum and its alloys, which constitute the greater part of the die-casting industry of today. The proper gating and venting of these dies are problems that arise daily and on the solution of these problems depends the success or failure of the process.

Alloys.—In a paper read before the American Institute of Metals in 1914, the writer described the various types of zinc, tin and lead alloys used in the die-casting process. The application of these alloys and their limitations were also pointed out. At that time the die casting of aluminum and its alloys was barely beyond the experimental stage. During the past four years the most important advance in the art of die casting has been made in the perfection of the process for die-casting aluminum and its alloys. The importance of this achievement as an aid to winning the war is best demonstrated by the fact that at least 95 per cent of the die-cast parts used directly or indirectly as materials of war were made from an aluminum-base alloy. Of these castings, only a very small percentage could have been produced successfully in 1914.

Investigations of the casting properties of metals and alloys in the past have been generally limited to sand castings; few data are available as to the casting properties of metals or alloys in metallic molds. Just what constitutes a good die-casting alloy is a subject of unusual interest. A few of the important requirements, outside of the usual physical properties demanded of alloys, are:

Melting-Point.—The successful die-casting machine in every instance is constructed of iron, in one form or another. The melting-point of the alloy must be such that it will melt readily in an iron pot.

Solvent Action.—The solvent action of the alloy on iron must not be too great. Molten aluminum dissolves iron very rapidly and analyses of aluminum die castings on the market will show an iron content of from 1 per cent to 3 per cent, due to the solvent action. Fortunately there is no serious objection to the presence of iron in aluminum casting alloys. Should the aluminum absorb much above 3 per cent iron, the melting point becomes too high and the alloy becomes viscous and unsuitable for making castings.

Elongation.—The elongation of the metal is of vital importance in determining the die-casting properties of an alloy. Not only is it desirable to know the elongation of the alloy when cold, but it is of greater importance to determine the elongation at various temperatures ranging from the melting-point of the alloy down to normal temperature. The reason for this becomes apparent when the physical phenomena of the die-casting process are considered. Let us assume that a ring, 12 inches (30.48 cm.) in diameter is to be die-cast in a metallic mold around a metallic core. As the molten metal strikes the mold it solidifies. Here a change of state occurs that is accompanied by a reduction in volume, commonly termed shrinkage. Unlike a sand core, the metallic core is not compressible, and retains its original size and form so that the shrinkage of the metal is converted into a stretching action on the solidified casting. If the elongation of the alloy at that temperature is not high enough to withstand this stress the casting will crack. In the usual die-casting practice it is not practical to remove the casting from the die at the solidification temperature of the alloy. For example, the solidification temperature of the aluminum-copper alloys used in the die-casting process is approxi-

mately 1,150 degrees F. (621 degrees C.). It has not been found practical to run the casting dies above a temperature of 500 degrees F. (260 degrees C.), which means that the castings are withdrawn from the dies at that temperature. It follows that the casting is subjected to another stretching stress after the casting has solidified, due to the contraction in volume that must occur when a casting is cooled from a temperature of 1,150 degrees F. to 500 degrees F.

The writer has been unable to find any reliable method for determining quantitatively the elongation of alloys at various temperatures. Many methods have been suggested, but they have proved of doubtful value. The simplest way is to use the old "try-and-see" method. To test the alloy, a casting is made in a die having a comparatively large core and thin wall. If the alloy can stand the casting stress, a perfect casting will be obtained, otherwise the casting will show bad cracks. Only a comparative result is obtained, but for everyday control it answers the purpose. However, a simple and reliable method for determining quantitatively the elongation of metals and alloys at various temperatures would prove of enormous value to all metallurgists engaged in the various phases of metal-casting research.

It is interesting to note that the elongation of a metal or alloy at normal temperatures is no indication as to the properties of that metal or alloy at higher temperatures. The writer has found many cases where an alloy showing little or no elongation at normal temperatures shows a high elongation at higher temperatures. The alloys of aluminum and copper may serve to illustrate this point. It is well known that the addition of copper to aluminum reduces the elongation of the aluminum alloy. An aluminum alloy containing 12 per cent copper will show less elongation than an alloy containing only 6 per cent copper when tetsed at normal temperature. Nevertheless, the 12 per cent copper alloy has a greater elongation at higher temperatures than the 6 per cent alloy, and consequently the 12 per cent alloy is better able to withstand the casting stresses to which it is subjected in the die-casting process.

In the early days of the die-casting industry alloys were compounded indiscriminately, and little or no consideration was given to the metallurgical principles involved. The manufacturer in many instances knew much more about machinery than about metals. The result was that there were put on the market die castings made from alloys that deteriorated rapidly and created a prejudice among engineers against the use of these castings. That this prejudice was in part justified must be admitted; nevertheless, the modern die-casting plant is equipped with physical and chemical testing laboratories and the die-casting practice of today bears no relation to that of five years ago.

Die Castings Made for War Purposes.—Die castings have had their most severe test during the past two years, during which time most of the die castings manufactured were used directly, or indirectly, in the Government's war program. Here is a partial list of the application of die castings for this purpose:

Gas masks, breather tubes and other metal parts.

Lewis machine guns, 100 die-cast parts to every gun.

Browning machine guns, four of the most vital parts.

Naval and army binoculars, the entire housing.

Army truck, tank and airplane die-cast parts include parts of ignition system, carburettor gasoline-regulating devices, steering-wheel accessories, ball-bearing cages bearings, speed indicators, &c.

Pistol, complete signal pistol.

Submersible bombs, some designs contained as many as 10 die-cast parts.

Hand and rifle grenades, every grenade manufactured in this country contained one or more die castings.

Trench mortar shells, plugs die cast.
Airplane drop bombs, one or more die-cast parts.
Surgical instruments, including hair clippers, respiratory devices, &c.

In many instances, die-cast parts were used where the failure of the part would result in serious loss of life. The fact that not one failure of a die casting has been reported must continue to be a source of deep satisfaction to the modern die-casting manufacturer.—"Engineering."

ON THE METALLURGICAL INFORMATION REQUIRED BY ENGINEERS.*

By Lieut.-Colonel C. F. Jenkin, R. A. F., Member.

For the past four years the writer has been responsible for recommending materials for all parts of aeroplanes and aero-engines. In engineering it has been customary to adhere to the use of materials which experience has shown to be suitable for the different parts, but in aircraft a new factor is of paramount importance—viz.: weight—with the result that new materials, such, for example, as aluminum alloys, are constantly being suggested and tried for purposes for which they have never before been used. Further, it has become of importance to use all materials in their strongest condition; thus the bulk of the steel used is so treated as to have a strength about double that of the steel used for ordinary engineering work. In order to be able to judge whether a new material—or an old material in a new condition—is suitable for any particular use the following fundamental problem therefore becomes of essential importance: "On what properties of the material does its suitability for any particular purpose depend?"

This question has arisen in connection with all sorts of materials—metals, timber, flax and hemp, rubber, paint, &c., and in connection with their use in engines, aeroplanes, balloons, armament, instruments, &c. To simplify the problem let us confine our attention to metals for use in engines. In order that a metal may be suitable for a particular part of an engine it must possess certain properties which the engineer can easily specify, *e.g.*:

1. Stability, *i.e.*, strength to resist deformation—beyond a minute extent.
2. Long life, *i.e.*, endurance when subjected to stress—of a definite sort—and when subjected to abrasion and corrosion.
3. Special properties for certain parts, *e.g.*, a low coefficient of friction for bearing metals.

For simplicity we may confine our attention to items 1 and 2. Stability and long life, then, are the properties desired by the engineer; but these are not among the properties which are known to the metallurgist, and when the engineer asks for information he is given a very different list of data, viz.: Chemical analysis; microstructure; mechanical tests—ultimate strength, elastic limit, elongation, reduction of area; bend tests of different sorts; Brinell, scleroscope; Izod; fatigue range; chemical reactions.

There is hardly any correspondence between the two lists. In an attempt to bridge the gap between the two we use a terrible list of vague words to hide our ignorance—Hardness; springness; toughness and brittleness; fibrous or crystalline structure; resistance to shock or vibration or impact.

Some comparison may be attempted between the two lists by putting them in parallel columns:

* Note read before the Institute of Metals, March 26th, 1919.

Stability under tensile or compressive stresses	Elastic limit.
Stability under bearing pressure (rivet holes, pin holes, ball bearings)	Scleroscopic hardness. Brinell hardness. Resistance to scratching.
Life under steady stress	Elastic limit.
Life under varying stress	Fatigue range.
Life under abrasion	"Hardness" (one of the meanings).
Life under corrosion	The chemical reactions on which corrosion depends are little understood.
Of no interest to the engineer	Chemical analysis.
Of no interest to the engineer	Microstructure.
Little interest except as giving a margin of strength before final fracture in some steadily stressed parts	Ultimate strength.
No interest	Elongation.
No interest	Reduction of area.
No interest	Bend test.
No interest	Brinell.
No interest	Izod.

To illustrate the want of correspondence between the two columns, let us suppose that it is desired to replace steel by an aluminum alloy for some purpose. It is obvious that it is impossible to make any useful comparison by comparing the analyses or the microstructure of the two metals. It is really useless to compare the ultimate strengths, elongations, reductions of area, Brinell, and Izod figures. For consider the elongation. Why do we specify 17 per cent elongation for a steel crank shaft when 1/10 per cent, if it occurred, would ruin it? Is there any reason, then, to specify 17 per cent elongation for an aluminum crankshaft? These apparently important mechanical tests are really no more rational guides than, say, color. As a matter of fact, an experienced eye can judge as well by the color of the steel fracture as by the Izod figure of a fracture. But it is obviously useless to compare steel and aluminum by color.

If the metallurgist's properties are so useless to the engineer, why are so many of them included in the specification? Owing to the lack of knowledge about the life and stability of materials it is not possible to specify such qualities, and the engineer has to specify some definite material which he wishes to use, either for an experiment or as a result of former experience. For this purpose he uses the chemical analysis to define the material, and the microstructure and mechanical tests to define the condition of the material.

What, then, is needed to enable a satisfactory answer to the problem to be given?

1. It is necessary to ascertain what physical properties of the metal they are which determine its behavior under the conditions specified by the engineer.

2. It is necessary to determine tests which shall show the extent to which those physical properties exist in any metal.

3. The names for these properties must be accurately defined.

As an example, consider a crank shaft. It is suggested that the only properties which may need to be considered are:

1. The fatigue range. The designed stress must not exceed this.

2. Freedom from sharp cracks or faults. Such faults may be due to casting flaws or to a crystalline structure which encourages surface cracks during machining.

Wearing property may be found to be unnecessary with properly lubricated bearings—if not, this must be added to the list.

All the other properties enumerated on the previous pages may be found to have no direct bearing on the suitability of a material for crankshafts. It may be gratifying to a steel maker to see a broken crankshaft twisted into all sorts of shapes after an accident, but why should it not have snapped off like glass?

The above two requirements for a crankshaft may be referred to more elementary properties by the metallurgist. He may be able to indicate the types of solution or crystalline aggregate which are likely to give the desired results. If we could arrive at such a result it would be a great advance on our present position. Such complete solutions have been reached in a few cases, and are certainly not impossible.

As an example of a test properly designed to determine the required property, Strohmeyer's test for the fatigue range may be given. A little further research on this test will probably make it perfectly trustworthy. There is no doubt what it measures.

A satisfactory answer to the problem should not be beyond the reach of the metallurgist and engineer if they work in conjunction. The reason that former attempts have had such relatively unsatisfactory results appears to be that engineers and metallurgists have not worked together, and that engineers have not fully realized the fundamental nature of the problems they were attacking.

It may be possible to correlate the essential engineering properties with the results given by existing types of test, and it is on these lines that most work has hitherto been done, but more probably new methods of testing will have to be devised. If new methods of testing are needed it is important that they should be designed to give the precise information required. Too many new tests have been devised in recent years without any exact knowledge of what their results signify. As a consequence a great amount of research has been expended in endeavoring to interpret the meaning of the new tests, which research would have been more profitably spent in devising tests to give the essential data.—"The Engineer."

INSTRUCTIONS FOR SAFE USE OF PULVERIZED FUEL.

Pulverized coal fuel involves certain elements of danger due to inflammable and explosive properties, as is the case also with oil and gas fuel. In order to ensure efficient service and to avoid accidents, therefore, it is necessary to provide careful handling by men properly instructed as to the conditions. Since accidents might tend to bring this comparatively modern system of fuel into disrepute a set of rules and instructions relating to the preparation, storage and use of pulverized coal have been prepared by the Pulverized Fuel Equipment Company, of New York, United States. These rules are given below, with some supplementary information from another company.

GENERAL INSTRUCTIONS FOR FUEL PREPARING, STORING, AND DISTRIBUTING EQUIPMENT.

NOTE.—Stop all fuel leaks.

1. The operation of pulverized fuel drying, grinding, distributing and storage plants should be under the supervision of employees who are

fully instructed and informed with respect to the proper care, upkeep and operation of all the equipment in conformity with the Pulverized Fuel Equipment Corporation's rules and regulations for the preparation, handling and burning of pulverized fuel.

2. Pulverized fuel in suspension is, under certain conditions, both inflammable and explosive. At each entrance to the fuel-preparing building a permanent sign reading as follows should be conspicuously placed:

"INFLAMMABLE—KEEP LIGHTS AND FIRE AWAY—SMOKING PROHIBITED—AVOID FUEL-DUST LEAKS."

3. A leakage of pulverised fuel is to be avoided the same as a leak of steam, gas, or fuel oil.

4. All bins, elevator casings, conveyor boxes, dryers, and pulverizers should be kept absolutely dusttight.

5. Wood or other combustible material must not be used under any circumstances in connection with bins or containers for wet, dry or pulverized fuel.

6. Accumulation of pulverized fuel on surfaces where it may be dislodged and become suspended in the air, or the formation of any dust cloud, is to be avoided, and the building and equipment for preparing the fuel should be kept clean at all times.

7. Fuel dust should be swept off the machinery and other surfaces, and not blown off with compressed air.

8. Only incandescent lamps equipped with heavy guards should be used for lighting in fuel-preparing plants. Where it is necessary to use a light in a confined space, such as an elevator casing, separator or bin, this should be done with great care, so as not to allow the lamp to be broken, and the bulb should never be allowed to lie against pulverized fuel in storage.

9. The use of matches, open fires, lanterns ,or torches about the pulverizing plant should not be allowed.

10. Smoking about the premises should be forbidden.

11. The temperature of the gases at the discharge end of the fuel dryer should not be allowed to exceed 400 degrees as registered by the pyrometer placed in the flue to the exhauster or stacks.

12. In starting fire in the dryer furnace use the smallest amount of waste or wood possible. Under no circumstances should shavings be used.

13. The use of air for the transfer of pulverized fuel from one container to another should be avoided.

14. In shutting down the plant at the end of a day's work all fuel should be fed from the dryer, the dried coal bins, and the pulverizer.

15. The National Board of Fire Underwriters* should be requested to pass upon any terminal arrangement put into effect for the handling of pulverized fuel.

SUPPLEMENTAL INSTRUCTIONS.

16. Local conditions may necessitate the establishing of additional instructions.

EMERGENCIES.

17. Should pulverized fuel in bulk ignite when in storage bins or containers other than those directly connected with the furnace combustion chamber, the containers should be flooded with water through inlets, letting this water with the coal pass through the outlets.

* Insurance authority..

18. **Should pulverized fuel in containers connected with the furnace combustion chamber become ignited, it should be fed to the furnace in the regular manner as quickly as possible.**

INSTRUCTIONS FOR FUEL FEEDING, BURNING, AND FURNACE EQUIPMENT.

1. Be sure the stack damper is open before the fuel feed is started.
2. Be sure feeder air is turned on full before the fuel feed is started.
3. Be sure fuel feed is entirely stopped before feeder air is shut off.
4. Fuel as delivered to pulverized fuel bins should have less than 1 per cent of moisture, and should be pulverized so that 82 per cent will pass through a 200-mesh sieve.
5. All bearings should be kept well lubricated, a good dynamo oil being used on motors and blower bearings and a good grade of cup grease on the other bearings.
6. Before firing the boiler be sure to open the stack damper to give about 1/10-inch draft in the furnace. Hang ignition fire of oily waste about 12 inches from the mouth of the burner, or build a wood fire on the floor of the mixing oven, according to the nature of the fuel that is being used.
7. Be sure that the feeder blower is up to speed, and that air at 6-inch water pressure is passing through the feeder and burner.
8. After making sure that the ignition fire is burning and full air blast is passing through the burner nozzle, start one feeder at about 25 revolutions per minute. Fuel should ignite at once. As soon as the fire begins to burn well, start the second feeder—this applies to mixing ovens having twin burners—at about the same speed as above.
9. Speed up the feeders gradually and adjust the auxiliary air openings and stack damper until the boiler is under the required load. Where two feeders are used their speeds should be about equal.
10. While the boilers are under load the only attention required is to maintain the water at correct level, keep the furnace free of ash, and clean the burner mouth and auxiliary air openings when necessary.
11. Blow off heating surfaces every six hours to maintain high efficiencies.
12. To put out the fire or bank the boiler, stop the fuel feeders completely, shut off air from the feeders and close the stack damper and auxiliary air openings. This will keep the boiler hot and protect the brickwork.

The Holbeck system of handling pulverized coal fuel differs from most other systems in that the pulverizing and storage facilities are at some distance from the factory or works containing the furnaces which are to be served. The Bonnot Company, of Canton, United States, which manufactures the apparatus for this system, has no general set of rules, but issues the following advice in regard to storage:

1. Moisture in the coal should be reduced to about 1 per cent before pulverizing.
2. Pulverized coal should not be stored in bins for periods to exceed three days.
3. Pulverized coal has a tendency to collect or absorb moisture, and if allowed to stand too long spontaneous combustion will result.
4. In case spontaneous combustion does not result, the coal will bridge or arch in the bin, thus making it difficult to feed it out for use.

In this Holbeck system the coal pulverized at the distant plant, as noted above, is delivered to the furnaces by means of a distribution main. The coal is fed from the storage bins to the main plant by means of screw conveyors discharging into the suction side of a blower. It is then conveyed to the furnaces in a non-explosive mixture of air, the ratio being about 60 cubic feet of air to 1 pound of the pulverized coal. The coal which is not used at the furnaces is then returned to the bin at the main storage plant to be used over again.

Under this system, instead of having separate bins for the several

furnaces, the pulverized coal stock is available for any furnace. There is also the comparative safety of having the storage bins at the pulverizing plant and away from the high temperature near the furnaces. Furthermore, there is said to be a minimum of danger of spontaneous combustion, owing to the fact that there is continuous movement of the pulverized coal from the top to the bottom of the bins, since the coal is fed out from the bottom while that not used at the furnace is returned and enters at the top of the bin.

The following sets of rules are issued by the Fuller Lehigh Company, of Fullerton, United States:

Pulverized coal is harmless unless mixed with air; therefore the following rules and regulations should be observed:

1. No smoking, open lights, torches or matches should be permitted around a coal pulverizing plant. Use a flash light or cage-protected incandescent light. The electric-light bulb should never come in contact with the fuel in storage.

2. Never attempt to enter any tank or bin that contains pulverized coal for any reason whatever. Never insert your head within for investigation without first thoroughly airing the interior. Pulverized coal-storage tanks may contain methane, an odorless gas, which will quickly render unconscious anyone breathing it. When necessary, tanks may be aired by opening the manhole doors. When a smolder or fire exists in tanks or bins they should be closed tight. This will tend to smother the fire due to lack of air to support combustion.

3. The pulverizing mill building should be well ventilated.

4. Storage bins, containers, or conduits must be kept airtight and free from pockets and leaks. Corners or projections should be avoided.

5. Surfaces where pulverized coal accumulates should be kept clean.

6. Do not open elevator casings, conveyor covers, or pulverizing mills while machinery is in operation. After shutting down the machinery allow sufficient time for coal in suspension to settle.

7. Do not hammer or make repairs on elevator casings or any other part of building while plant is in operation.

8. Do not use waste to stop dust leaks.

9. Keep all elevator casings and conveyor troughs tight, and keep the entire building as dustless as possible.

10. Make daily inspections.

11. When the plant starts, run sufficient non-combustible material through the equipment to fill all spaces where coal might lodge. Do this occasionally in order to keep such spaces filled.

12. Always keep bottom of gear casing and pulleys or belt guards clean, or, if possible, construct them in such manner that no oil refuse can accumulate or catch fire from friction or moving parts.

13. Be particular concerning cleanliness.

14. Safety devices should be properly installed for protection from accidents around all machinery.

15. When starting up a furnace using pulverized coal as a fuel, open the damper, build the fire, turn on the air and then the coal. When shutting down turn off the coal and then the air. Failure to do this may lead to a puff or back-fire.

16. Never attempt to ignite the mixture of air and coal in the furnace by reaching in or opening the doors.—"The Engineer."

ABRASIVES FOR SPECIFIC USES.

By R. G. Williams.

The manufactured emery wheel was a decided advantage over the natural sandstone, but as manufacturing conditions grew more exacting, an attempt was made to produce a wheel which would more nearly meet

the requirements. Corundum belongs to the same family as emery, but is purer. This material was more satisfactory than emery, but because the quantity was limited and because it was difficult to obtain succeeding shipments of uniform quality, even corundum was not entirely satisfactory.

The electric furnace has made it possible now to use abrasives which are constant in quality and practically unlimited in quantity. Two distinct classes of electric-furnace abrasives are in existence, one known as the aluminous group, containing aluminum oxide as the base, while the other is known as the carbide-of-silicon group, and is marketed under certain trade names as crystolon, carborundum, xolon, etc. These electric-furnace products are known as artificial abrasives.

There is hardly an article of commerce which does not come in contact with abrasives in at least some stage of its manufacture.

The following list is comprehensive enough to give a conception of the forms and classes into which abrasives can be divided:

1. Grindstones, pulpstones, and millstones.
2. Infusorial earth, Tripoli, rottenstone, pumice, etc.
3. Garnet, quartz, chilled steel.
4. Hones, oilstones, and whetstones.
5. Diamond dust and bort.
6. Emery and corundum.
7. Artificial abrasives of two classes—namely, aluminous and carbide of silicon.

Sandstone wears more rapidly on any grinding operation than would a wheel containing an artificial abrasive, but since sandstone costs much less than a modern grinding wheel, it requires consideration of wear and cost to determine which is the more economical.

Artificial abrasives in themselves have no effect upon the human system, in that the various human fluids have no chemical or physiological action on the abrasive. However, no doctor would approve a man working in an atmosphere containing a large amount of abrasive dust. The irritation which would result in the nose, throat and lungs would in turn produce a situation whereby such diseases as tuberculosis might develop. The dust from all dry-grinding operations should be removed. Just why sandstone should cause lung diseases to develop rapidly has not been determined. Nevertheless, it is a fact that even uneducated operators are forced to give up work which requires the use of sandstone, after a year to a year and a half.

Infusorial earth, Tripoli, rottenstone and pumice are comparatively soft abrasives, and readily break down into the form of a fine powder, so that when used with water a mud is formed which aids in polishing.

Garnet is also a natural mineral, very much softer than either emery or corundum. On account of its chemical composition it cannot be bonded in the form of a vitrified wheel, but must be used in the form of layers or strips held to a background by means of glue or cement.

The nature of garnet makes it particularly well adapted for working on wood, and garnet discs find extensive use in the pattern shop for this reason.

Quartz is used mostly for grinding glass, and ordinary sand of good quality is the form in which it finds commercial use.

The action of abrasives when used in the form of hones, oilstones and whetstones differs from that when used in the form of wheels in that the cutting action is slow. As a very keen, sharp edge is essential to the metal being cut, it is therefore necessary that the abrasive be of small size. The stones are used either with water or oil, or, in rare instances, dry. Stones containing artificial abrasives have largely replaced natural stones.

In the form of dust, diamond and bort are used for lapping, or for cutting and polishing gems. A lap consists of soft metal, such as copper

or cast-iron, in which is imbedded small particles of the diamond dust or bort. The lap is revolved, rotated, or reciprocated at slow speed, and is used only where very high refinement is required.

Emery is a natural mineral. At present the mines producing the greatest tonnage are in Turkey and Greece. It is a mechanical mixture of crystalline aluminum oxide and magnetic oxide of iron, the ratio being, roughly, 60 to 65 per cent aluminum oxide and 40 to 35 per cent magnetic oxide of iron. Since magnetic oxide of iron has very low abrasive properties, emery is not an efficient abrasive, in that only approximately two-thirds of it is of such a nature as to cut effectively. Due to the fact, however, that it does contain so much magnetic oxide of iron, it bonds very satisfactorily, and becomes extremely hard in the form of vitrified grinding wheels.

Corundum is a purified form of emery, in that it is 80 to 85 per cent crystalline aluminum oxide. A very good grade of corundum was discovered in Canada, and placed on the market shortly after 1900. The supply is limited, and as the artificial abrasives have come into use, Canadian corundum has diminished. There is a good grade of corundum in India, but not an important factor in the American market.

There are various artificial aluminous abrasives on the market, all differing somewhat in both physical properties and chemical composition. The common trade names are alundum, aloxite, boro carbon, adamite, etc. They consist essentially of 90 per cent or more of aluminum oxide, the remaining materials being known as impurities, and consisting of iron oxide, titanium oxide, and silica. Abrasives of this class are produced in the electric arc furnace. Bauxite is the most satisfactory raw material to use, although emery is also used. The manufacture consists in fusing a material high in aluminum oxide under the high temperature of the electric arc, with a reduction of the impurities. The abrasive floats on top as slag, while the impurities in a very low state of oxidation settle to the bottom, and are later separated by mechanical means and a magnetic separator after the furnace operation is completed. After the current has been shut off from the furnace the pigs are allowed to cool for a number of days before being broken up into small pieces. These small pieces are further reduced in size by powerful crushers and rolls into sizes varying all the way from pieces approximately 1/10 inch across, to particles so fine that they will stay suspended in water for a long time.

All carbide-of-silicon abrasives are artificial, and also depend upon the electric furnace for their manufacture. The raw materials are coke, sand, sawdust and a little salt. The resistance furnace is used, and a cross-sectional view would show a central core of granular carbon, around which is placed a mixture of the coke, sand, sawdust and salt. The heat caused by an electric current passing through the carbon core is sufficient to result in a union of the carbon in the coke and the silica in the sand, producing a compound containing one atom each of carbon and silicon.

Pure carbide of silicon is probably colorless. The purest form known is light green, while the commercial varieties vary from green to a product dark blue to jet black in color. The chemical composition of the different kinds varies but slightly, but this difference, together with differences in furnace treatment, produces abrasives which vary enough in their physical properties to affect their commercial uses.

Numbers are used to designate the different sizes of abrasives. Thus a No. 24 abrasive means that that particular size of grain has passed through a 24-mesh screen, but is too large to pass through the next finer screen, which has 30 holes to the linear inch. Sizes finer than 200 are designated by letters, and are referred to as flours, because the particles are so fine that it is hardly accurate to refer to them as grain.

Coarse grains are used for operations where the rapid removal of stock is necessary, while the finer sizes are used in wheels intended to remove

a small amount of material and produce a good finish on the work. The flours are used in very fine oilstones, rubbing bricks, etc.

Mention should be made of the use of abrasives for polishing purposes. For this work the abrasive is not bonded in the form of a solid wheel, but is held to suitable wheels of canvas, leather, etc., by means of glue.

The object in polishing is not to remove material rapidly, but rather to take small shallow cuts, so that eventually the surface presents an attractive appearance. Aluminous abrasives are used mostly for polishing. If carbide of silicon could be readily held to a polishing wheel by means of glue, this abrasive would be used more extensively than it is.

Hardness is undoubtedly the most influential property of an abrasive. Mineralogists use what is known as Mohs' scale of hardness, in which the diamond receives a value of 10, corundum is numbered at 8, while topaz is numbered at 8. This scale is undoubtedly misleading, in that there is a much greater difference in hardness between diamond and corundum than corundum and topaz. Nevertheless it is in common use, principally because materials harder than corundum present great difficulty in being properly tested for hardness. Tests have been made indicating that the artificial aluminous abrasives are considerably harder than corundum, though not nearly so hard as diamond, while carbide of silicon is between the diamond and the artificial aluminous abrasive in hardness.

Next to hardness the most important physical property of an abrasive is toughness. Toughness is very desirable. While the individual cuts of each abrasive particle are small, nevertheless they are deep enough so that considerable resistance is brought on the grain, which, to be efficient, particularly in cutting materials of high tensile strength, must possess the property of toughness.

The reason the two types of artificial abrasives, aluminous and carbide of silicon, are in such extensive use, is primarily because carbide of silicon, while extremely hard, is not so tough as the aluminous abrasives, which in turn possess less hardness than carbide of silicon, but are much tougher. Practical experience extending over a period of years has proven that for materials of high tensile strength the aluminous abrasives are best adapted, while for materials of low tensile strength carbide of silicon is most efficient. The meeting point is probably at malleable iron. For materials of less tensile strength than malleable iron, such as brass and bronze, granite and marble, and leather and wood, it will be found that carbide of silicon will produce more satisfactory results than will an aluminous abrasive; while for steels in general an aluminous abrasive should be used for satisfactory results.—"Industrial Management."

FLAKES IN ALLOY STEELS.

The occurrence of "flakes" in "tender" nickel-steels and nickel-chrome-steels, leading to the rejection of forgings of these alloys, was one of the unpleasant experiences made during the war, especially in the United States. Cracks would develop suddenly, without apparent reason, in such steels when under test, and could not always be cured by heat treatment. The American Institute of Mining Engineers discussed the question recently in connection with papers by C. Y. Clayton, F. B. Foley, and F. B. Laney, and by H. S. Rawdon, and H. M. Howe presented a report on behalf of the Steel Ingot Committee of the National Research Committee. These papers were noticed in *Metallurgical and Chemical Engineering* of March 1 last, and a further instructive communication on the subject was contributed to the March 15 issue of the same journal by Frederico Giolotti, the Italian metallurgist, who was in New York at the time. The flakes, also known as snow-flakes, scabs, goose eggs, silver streaks, commonly appear as bright streaks, sometimes greenish in the

central portion, in alloy steels which show weakness when worked. The defect is brought out in the macrostructure by etching; radiographs show white lines; micrographs indicate lack of grain refinement. The flakes or streaks are supposed to be cracks in the steel itself; they are not found in steel which welds readily, but are troublesome in nickel-steels which do not weld. Clayton and his collaborators consider that there is always evidence of overheating in flaky steel, and they adduce a good deal of experimental evidence favoring his view; the flakes, they say, pass through the ferrite existing between sorbitic grains, and also through the grains themselves. Rawdon believes that the flakes represent original crystal faces which formed during the solidification of the inhomogeneous alloy steel; the rupture would take place already in the ingot owing to the different expansion coefficients of the metals and alloys. Howe inclines to the latter view. Flakes, he points out, also appear in alloy steel which has not been heated above 900 degrees C., and are not characteristic of the zones which were hottest in the ingot. They vanish in some steels, as the nickel percentage is reduced, he admits, whilst they increase in alloy steel which has been killed in the furnace. Tender steel should hence be humored and so cast that excessive cooling be avoided, using round molds, stripping as soon as the walls are strong and cooling by burying in ashes. This treatment would tend to favor dendritic structure, but it would lessen the flaking. In their *résumé* Howe and Rawdon agree that flakes originate in the ingots of tender alloys as intercrystalline shrinkage cracks and are probably intensified by inclusions of segregations. Giolotti would accept this conclusion; he would say that the flakes originate in tender alloys as intercrystalline cracks, but he does not believe that they are formed in the ingot. He remarks that the Ansaldo Company, with which he is connected, made 10,000 guns for the Italian Army, and had no rejection on account of flakes, though they were troubled with them. His chief point is: to prevent flaking, keep not only sulphur and phosphorus as low as possible, but avoid all iron oxide, do not use rusty turnings or borings, and fuse in a reducing atmosphere under a thin covering of silica slag. He adds the ferrochrome 20 minutes before pouring (hardly any chrome should in good practice go into the slag) and pours hot and cools in the air—contrary to Howe; he has not observed any preferential directions of the flakes in molds of various shapes (Howe). He ascribes the trouble to the diffusion of emulsified, microscopically undetectable iron oxide through the metal. That non-metallic inclusions are surrounded by decarbonized zones he explains by the presence of both CO and CO_2 in the steel (dissolved, not in cavities); the equilibrium: $3Fe + 2CO \rightleftarrows Fe_3C + CO_2$ is shifted to the left by an oxidizing slag so that a zone of decarbonized iron is produced, supersaturated with CO. There is no internal crack, however, until the material is strained above the elastic limit.—"Engineering."

THE TONNAGE OF MODERN STEAMSHIPS.*

By A. T. Wall, A. R. C. Sc., Member.

1. The main object of this paper is to focus attention on the effect of recent legislation and modern machinery on tonnage measurement. Some remarks on the basis of measurement are also made.

2. *Basis of Measurement.*—It will not be out of place to touch on the principle underlying the whole question, especially as it is thought by many that the present method is not only complicated but unfair as between ship and ship. The subject is obviously quite unsuitable for exact mathematical

* Paper read before the Institution of Naval Architects, April 11, 1919.

treatment, and it must be recognized that whatever basis is used, anomalies are bound to occur.

3. This side of the matter has been dealt with by Mr. E. R. Johnson in his book on "Measurement of Vessels for the Panama Canal." He gives several bases that have been proposed for tonnage measurement, and considers them in a general way in coming to the conclusion that the only fair and practicable method is the present one of assessing the earning spaces in the ship. The measurement of vessels for the Panama Canal is very similar to that under the Suez Canal Rules, which is the same in principle in all maritime countries.

4. The proposed bases are as follows:

(1) Gross deadweight.
(2) Cargo deadweight.
(3) Block displacement (length × beam × draught).
(4) Displacement.

5. Gross deadweight is obviously an unfair method, as it takes no account of the passengers carried in the ship usually having a small deadweight.

6. Cargo deadweight would be very difficult if not impracticable to assess, and, to be logical, would vary under different conditions, requiring repeated measurement. The same objection as to passengers applies as in paragraph 5. Further, ships carrying no cargo would have no tonnage.

7. Displacement tonnage would appear to be fairer than the foregoing. This tonnage is made up of the light weight of the ship and the gross deadweight. The former is heavier in a passenger ship than in a cargo vessel of the same length for many reasons; among these are greater superstructure and heavier machinery. In addition, large quantities of fresh water, stores, baggage, &c., must be carried for the passengers. This excess weight gives a measure of the passengers carried.

8. A great advantage of displacement tonnage would be that it is easily measured, and a certified displacement scale would be carried by the ship. The tonnage on this basis would also vary with the loading of the ship and so be more commensurate with her earning capacity and the services rendered to her for which dues are levied.

9. Block displacement tonnage is similar to displacement tonnage, but is severe on vessels of fine form.

10. It would be impossible by any system to take an accurate measure of the value of the cargo and the revenue from passengers, even if this were a proper basis for tonnage dues, which is doubtful. It would be equally impossible to measure properly the actual service performed by any port, canal, &c., to a particular ship. Any basis must be somewhat arbitrary, but the existing method does, or at least could, be made to give a true measure of the earning spaces in a ship.

11. Table I gives particulars of various vessels, showing how the bases compare.

12. From Table I it is evident that no two bases will give corresponding results; but it shows that there might be some justification for a displacement basis or a block displacement basis.

13. *International Tonnage.*—Although the principle of measuring the actual earning spaces of a ship is adopted in all maritime countries, there are many variations in details which give the same ship very different tonnages under different rules. Gross tonnages do not vary greatly except on account of the exemption for shelter-deck and certain superstructure spaces as allowed under British rules. This causes very different values, for instance, between American and British measurements. One example is sufficient as an illustration. A British ship, with shelter-deck spaces

exempt, has a gross tonnage of 4,435, whereas under American rules it is 5,470, an increase of 23 per cent. The German rules make the same allowance as the British. It is often maintained that there is no justification

TABLE I.

No. of Ship.	Moulded Dimensions.	Draught.	Speed.	Total Number of Passengers.	Net Tonnage.	Gross Deadweight.		Cargo Deadweight.		Block Displacement.*		Load Displacement.	
						Tons.	Ratio to Net Tonnage.	Tons.	Ratio to Net Tonnage.	Cubic feet, ÷ 100.	Ratio to Net Tonnage.	Tons.	Ratio to Net Tonnage.
		ft. in.	knots.										
1	235′ × 34′ × 17′ 3″	15 11	8½	None	782	1,660	2·12	1,270	1·625	1,272	1·625	2,760	3·52
2	360′ × 45′ × 23′ 6″	20 11	14	116	1,835	3,135	1·71	2,230	1·215	2,827	1·54	5,928	3·23
3	312′ × 40′ 9″ × 26′	22 9	9	8	1,859	4,175	2·245	3,360	1·81	2,893	1·555	6,035	3·245
4	345′ × 43′ 9″ × 27′	23 1	11½	430	2,166	4,065	1·875	2,500	1·115	3,490	1·61	7,027	3·245
5	375′ × 47′ 3″ × 29′	24 4½	14½	519	2,497	4,270	1·71	2,700	1·08	4,319	1·73	7,990	3·20
6	376′ × 50′ × 34′	25 8	11	146	3,512	7,450	2·12	5,640	1·61	4,826	1·37	10,970	3·12
7	390′ × 51′ 3″ × 29′	23 11	11	None	3,110	7,386	2·375	6,200	1·995	4,781	1·54	10,353	3·34
8	400′ × 49′ 9″ × 30′	25 7	13½	517	3,223	5,780	1·79	4,060	1·26	5,057	1·57	10,125	3·14
9	400′ × 52′ × 31′	25 2	11½	None	3,178	8,180	2·575	7,280	2·29	5,138	1·615	11,485	3·61
10	410′ × 53′ × 30′ 6″	24 5	11	None	3,210	8,260	2·57	7,356	2·29	5,300	1·65	11,520	3·59
11	530′ × 64′ × 40′ 6″	31 6	17	1,800	7,598	11,750	1·55	7,700	1·015	10,685	1·405	21,590	2·84
12	530′ × 66′ × 45′	34 0	15	650	8,720	16,340	1·87	9,940	1·14	11,890	1·365	25,940	2·975
13	570′ × 69′ × 42′ 3″	31 7	11	None	9,780	19,815	2·025	18,362	1·88	12,420	1·27	27,330	2·795
14	600′ × 73′ × 45′	35 6	16	2,400	12,920	18,790	1·455	14,700	1·14	15,565	1·205	32,810	2·54
	Average value of ratio to net tonnage				..	—	2·00	—	1·58	—	1·51	—	3·19
	Mean percentage variation from the average ratio				..	—	14·5	—	25·8	—	8·5	—	7·8

* Length × breadth × draught.

for the exemption of the shelter 'tween-deck spaces. The tonnage laws of the American, Suez and Panama Canal authorities do not recognize it.

14. The allowances for net tonnage are very varied. Taking the same ship as in paragraph 13, her tonnages are as follows:

	Gross.	Net.	Percentage Increase Net Tonnage over British.
British	4,435	2,808	—
American	5,470	3,512	25
Suez Canal	5,478	4,203	50

15. Table II, taken from the book by Mr. Johnson already referred to, shows how greatly the deduction for net tonnage varies under different rules.

16. The disadvantages of varying rules are to some extent overcome by agreements between countries to take one another's figures, but in many cases additions are made, especially where the tonnage is much less than it would be if measured by the country collecting the dues.

17. The International Tonnage Commission, which met at Constantinople in 1873, drafted rules for the Suez Canal Company which it expected would have been adopted internationally.

18. The chief causes of variation in gross tonnage are the exemption of shelter-deck and superstructure spaces, while the particular rule adopted for propelling space deduction has most effect on the net tonnage. There are, however, many other differences which cannot be justified.

19. International rules should be instituted, and, if taken up in the right spirit, would not be difficult to bring about, especially from a technical standpoint.

20. *Tonnage and Freeboard.*—The existing Tables of Freeboard take account of the form of the vessel by using the tonnage coefficient. It is stated in the tables that this coefficient bears a constant relation to the whole external volume of the ship below the upper surface of the deck divided by the product of length × breadth × depth. This tonnage coefficient was adopted in the Tables of Freeboard to make the tables easily and directly applicable in cases where a displacement scale for a vessel is not at hand. In practice, for new ships, the tonnage coefficient is deduced from the displacement coefficient which is the only thing known in the design stage, so that the method adopted by the Committee on Load Lines in their report issued in 1916 (where they use the molded displacement coefficient at a draught which is 85 per cent of the molded depth to the freeboard deck), is much simpler, and will be welcomed when the report referred to becomes law, as it is understood will be the case.

21. There are also minor objections to the use of the tonnage coefficient for determining freeboard. In the first place, there is a good deal of work involved in correcting the tonnage coefficient for the standard sheer and round of beam, and also for the type of bottom construction adopted.

22. Further, this tonnage coefficient is intended to take account of the fineness of the ship up to the freeboard deck. The freeboard deck may, or may not, be the tonnage deck, and the position of the tonnage deck will depend on the number of complete decks carried, it being the second complete deck from the bottom. There will, accordingly, be a small variation in freeboard with the number of complete decks carried, which should not be the case.

23. *Tonnage and Subdivision.*—The exemption of shelter 'tween-deck and other superstructure spaces has been referred to in paragraph 13. It is doubtful, in passenger ships, owing to the present subdivision rules, whether it would be possible to exempt many such spaces.

24. To be exempt from tonnage, a shelter 'tween-deck must have a tonnage opening, openings in the transverse bulkheads, and scuppers discharging overboard from the deck under the shelter deck, that is, the upper deck.

25. The freeboard of such vessels brings the load draught only a little below the upper deck. If the upper deck is retained as the bulkhead deck it generally becomes necessary to put in so many bulkheads to meet the subdivision requirements as to make the ship impracticable. The alternative is to carry the watertight bulkheads up to the shelter deck. The necessary tonnage openings in these bulkheads cannot then be fitted, and the scuppers in the upper deck must also be omitted. This matter was discussed in detail by the author in a paper read before this Institution in 1916 on "Some Effects of the Bulkhead Committee's Rules in Practice."

26. The inclusion of the shelter 'tween-decks in the tonnage will increase the gross and the net tonnages. If the propelling space is still large enough with the light and air spaces to be over 13 per cent of the new gross, the increase in net tonnage will be of the order of 25 per cent. It may be, however, that the propelling space may not be large enough to be over 13 per cent of the gross, when the increase in net tonnage may be as much as 50 per cent. This is a very serious disadvantage of the subdivision rules.

27. One other case of a similar nature may be briefly mentioned. It occurs mainly with home trade passenger and cargo steamers of the three-island type. To subdivide adequately such vessels it will be necessary to fill in one or both of the wells, making the vessel flush deck forward or throughout. This adds to the tonnage, and although the wells are used to carry cargo on occasions, they are not always included in the tonnage measurement. An example of this is given in the paper referred to in paragraph 25, and the increase of net tonnage may be about 20 per cent.

28. *Deduction for Propelling Space.*—The British law, for propelling space allowance in screw steamers provides that when the tonnage of the space solely occupied by and necessary for the proper working of the boilers and machinery is above 13 per cent and under 20 per cent of the gross tonnage, the deduction shall be 32 one-hundredths of the gross tonnage. If the space does not lie between these limits the deduction is 1¾ times the actual space.

29. This rule was adopted in 1854, when tank boilers and reciprocating engines were the principal power units and coal the main fuel. Since then, and especially during the last ten years, great advances have been made with propelling machinery. Modern machinery, such as geared turbines, water-tube boilers, oil engines, electric transmission and oil-fired boilers, occupies much less space for the same power than the old machinery. Consequently modern machinery often occupies less than 13 per cent of the gross tonnage.

30. The result is a considerable increase in net tonnage, quite out of proportion with the increase in cargo space, as evidenced by the following example:

Ship with Old Machinery.

Ship dimensions (molded)	580 ft. by 70 ft. by 45 ft.	
Gross tonnage		20,000
Deductions for crew spaces, &c.	1,350	
Propelling space with coal-fired cylindrical boilers and Q.E. reciprocating engines	2,800 (14 per cent of the gross)	
Deduction on account of propelling space (32 per cent gross)	6,400	
Total deductions		7,750
Net tonnage with old machinery		12,250

TABLE II.—*Gross and Net Tonnages of the Metal Steam Vessels of the World*, 1910, *and the Percentage Deducted from the Gross in Ascertaining the Net.*

Flag.	Gross.	Net.	Per Cent. Deducted.
Argentinian	139,346	82,418	40
Austro-Hungarian ..	777,240	485,675	37
Belgian	295,918	194,336	34
Brazilian	232,425	142,582	38
British	17,940,862	10,893,898	39
Chilean	114,067	72,608	36
Chinese	84,800	54,526	35
Cuban	52,494	32,165	38
Danish	668,836	391,788	41
Dutch	982,104	607,286	38
French	1,445,422	835,016	42
German	3,959,147	2,416,370	39
Greek	498,281	312,296	37
Haitian	3,387	2,017	40
Italian	985,716	597,640	39
Japanese	1,064,169	675,983	36
Mexican	27,324	16,513	39
Norwegian	1,385,631	838,320	37
Peruvian	10,371	5,354	48
Portuguese	78,829	48,677	38
Roumanian	31,688	16,690	47
Russian	687,231	400,761	41
Sarawak..	3,953	2,380	39
Siamese	12,607	7,792	38
Spanish	746,047	459,198	38
Swedish	756,909	449,872	40
Turkish	110,770	68,753	37
United States (excluding Northern Lakes)	1,439,911	939,505	34
Uruguayan	48,387	30,118	37
Venezuelan	3,166	1,856	41
Other flags	20,657	11,869	42
Philippine	32,287	20,834	35
Total (national) ..	34,639,927	21,115,093	39
Suez Canal	23,054,901	16,581,898	28·07

TABLE III.

Type of Ship.	Type of Machinery	Percentage Reduction in Propelling Space on Space Occupied by Old Machinery.
1. 600-ft. intermediate passenger and cargo steamer	Coal-fired cylindrical boilers and double reduction geared turbines	13
2. ,, ,,	Oil-fired water-tube boilers and double reduction geared turbines	33
3. 530-ft. intermediate passenger and cargo steamer	Coal-fired cylindrical boilers and double reducton geared turbines	10
4. 400-ft. cargo steamer	Camellaird-Fullager oil engines	37
5. 460-ft. cargo steamer	Coal-fired cylindrical boilers and Ljungström turbo-electric engines	10
6. 150-ft. coaster ..	Camellaird-Fullager oil engines	20

Ship with Modern Machinery.

Gross tonnage (as before)		20,000
Deduction for crew spaces, &c. (say, as before)	1,350	
Propelling space with oil-fired water-tube boilers and double-reduction geared turbines (10 per cent of gross)		
Deduction on account of propelling space, 10 per cent by 1¾ per cent of gross	3,500	
Total deductions		4,850
Net tonnage with modern machinery		15,150
Increase in net tonnage	24 per cent.	
Increase in cargo space	5 per cent.	

31. The propelling space with the old type of machinery may be so little above 13 per cent of the gross tonnage that the substitution of geared turbines of the same power for reciprocating engines, and retaining the coal-fired cylindrical boiler, decreases the engine-room space and the boiler-room space so much as to bring the propelling space below 13 per cent of the gross tonnage. Actual cases of this have come under notice.

32. Some actual cases of the percentage reduction in propelling spaces are given in Table III.

33. The progressive shipowner who adopts modern machinery will therefore generally lose money by paying high tonnage dues, abroad as well as at home, or by deliberately making unnecessarily large machinery and light and air spaces and so wasting cargo and passenger space. The latter course may not be successful, as the Board of Trade have power to refuse to allow space in the measurement if it is not reasonable in extent for the intended purpose.

34. There is, of course, some gain in cargo space to set off against the increase in net tonnage, but it is not proportional. Examination shows that the result is a net loss. In case 2, Table III, the gain in cargo space is 6 per cent and increase in net tonnage 26 per cent. The annual loss in revenue has been estimated to be 4,000*l.*

35. The present rule fixed 13 per cent to provide comfort for those who had to look after the machinery, and the 32 per cent reduction was made to compensate for bunkers. There does not appear to be any justification for this percentage rule, and in view of its unfair incidence in many modern vessels, the question naturally arises as to whether it ought not to be dropped, and replaced by a rule which gives some multiple of the propelling space as allowance for bunkers.

36. To say that, as yet, few ships are affected is not strictly true, and even if it were so legislation should anticipate progress and help it.

37. As the British tonnage regulations have been adopted by a large number of nations, alterations would probably have to be made by international agreement, and the British law now regulating tonnage measurement would have to be altered by Act of Parliament. In the meantime a proposal is put forward for meeting the case. It is not thought fair that ships with propelling spaces not exceeding 13 per cent of the gross tonnage should have the 32 per cent deduction. To make matters more equitable it would appear reasonable to make the deduction the same percentage of 32 per cent as the actual propelling space is of 13 per cent. For example, a ship with 10 per cent space would have a deduction of $10/13 \times 32$, that is 24.6 per cent instead of 17.5 per cent under the present rules. Under this proposed rule the ship in case 2, Table III, would then lose practically nothing as compared with the same vessel fitted with the old type of machinery.—" Engineering."

STANDARDIZATION OF MARINE BOILERS.

The movement first initiated on the North-East Coast towards the standardization of marine engines, boilers and accessories led ultimately to the formation of a National Committee, to which was delegated the duty of considering the question in all its bearings. The committee, which is composed of members appointed by the Institution of Naval Architects, North-East Coast Institution of Engineers and Shipbuilders, Institution of Shipbuilders and Engineers in Scotland, Liverpool Engineering Society, Institute of Marine Engineers, and on which Lloyd's Register, the British Corporation, the Bureau Veritas, the Board of Trade, the Barrow and Belfast districts, as well as steel manufacturers, are represented, has been at work for some time past. Mr. A. E. Seaton has been chairman, and Mr. J. T. Milton has acted as vice-chairman.

The first subject which it was decided to investigate is that of marine boilers and accessories, and a report has now been prepared which deals with main boilers of the cylindrical type, auxiliary and special boilers, and with steam feed and other pipes subject to internal pressure. The opinion is expressed that at present it would serve no useful end to attempt a standardization of boiler design. The committee thinks, however, that standardization should be at once carried out in respect of the various component parts, whereby economy in their manufacture might be affected without placing restriction on general design. Special consideration has been given to the practice of the British Admiralty and the Naval and Mercantile Authorities which control boiler construction in foriegn countries. The committee has gathered a considerable amount of valuable evidence respecting the behavior of boilers of the type reported upon when working under service conditions, and has ascertained which parts are most liable to give rise to trouble. It has had some experiments carried out with a view to elucidating some points on which sufficient evidence was not forthcoming, and is having further experiments made with the same object, so that the evidence on which conclusions are based may be convincing.

The committee is convinced that basic open-hearth steel as now produced may be used with advantage in the construction of any part of a marine boiler, the tests and conditions imposed being the same as those on acid open-hearth steel. It considers that in a general way the standard 28-32 tons tensile steel should be used for the cylindrical shell plates and long stays; and, if required, steel having an ultimate tensile strength of 35 tons may be used, but beyond 35 tons it shall be a matter of arrangement with the classifying authority, the owners and builders.

The committee is of the opinion that the steel now supplied for boilers is so good and reliable, and that boiler design and methods of manufacture are so much improved, that they command much more confidence than formerly obtained. It finds also that boilers made for the Mercantile Marine in accordance with the rules of Lloyd's Register, and that the cylindrical boilers made under Admiralty rules, which permit even lighter scantlings than do Lloyd's, have proved by many years' experience to be safe and reliable. It, therefore, has come to the conclusion that the shell plates need be no thicker than required by the rules of Lloyd's Register, and so have made the formula for thickness of cylindrical plates on this hypothesis. Seeing that all boilers are now made entirely of tested material from designs made in accordance with approved rules for their safety, and under inspection during construction, there is no need to test them by hydraulic pressure to the extent that was formerly necessary.

Two of the most important points which are raised in the report, and on which the committee is at variance with the authorities, are the hydraulic test pressure, and the factor of safety for shell plates. Marine

engineers are aware that while the Admiralty hydraulic test only calls for a margin of 90 pounds over the working pressure, the Board of Trade has insisted on this test being made at double the working pressure. The variations in this respect are indeed somewhat remarkable, the French authorities requiring 85 pounds, the Germans 75 pounds and the Italians 71 pounds in excess of the working pressure for testing purposes. The differences in practice may be illustrated in another way. A North-East Coast standard boiler for a cargo vessel which is designed to work at 180 pounds pressure must pass a Board of Trade survey at 300 pounds pressure, whereas a French boiler would be tested at 265 pounds, a German-made boiler at 255 pounds and one made by an Italian firm at 251 pounds. An Admiralty transport boiler would have to undergo a test of 270 pounds. In the case of the shell plates a boiler 17 feet in diameter, having plates 1 inch thickness, would be passed by the British Admiralty for 190 pounds working pressure, by Lloyd's for only 154 pounds, and by the British Corporation for 157 pounds. It was the existence of these anomalies that led the committee to draw up a new rule for boiler-shell plates. Under this rule a boiler which is to work at 157 pounds pressure will be tested at 285 pounds hydraulic pressure, as compared with the 280 pounds pressure, which is the existing requirement of the Board of Trade and the Admiralty, although the former authority would only allow 140 pounds working pressure. The action taken by the committee is based on the belief that every useful purpose will be served if the hydraulic test to twice the working pressure is limited to boilers, the working pressure which does not exceed 100 pounds per square inch, and that for higher working pressures it will be sufficient if the test pressure is 50 pounds in excess of 1½ times the working pressure. This will mean that the stress at hydraulic test will be less on the scantlings than was the case with boilers having a factor of safety of 4.5. In view of the fact that manufacturers of plates prefer that thicknesses should be expressed in 32nds of an inch, rules involving thicknesses of plates have been based on that condition.

After consultation with tube makers the committee has drafted a specification which will be worked to by it in future. This step is necessary, as no test conditions have hitherto been laid down for the smoke tubes of boilers or even for the stay tubes.—"The Steamship."

JAMES WATT CENTENARY.

On August 25th of this year one hundred years will have elapsed since the death of James Watt, and it is proposed to commemorate the event in a fitting manner. A provisional scheme has been drawn up and will be submitted for consideration at a meeting in the Council House, Birmingham, at 2.30 P. M. on Thursday next, May 8th. This scheme provides for a celebration with a banquet and addresses on August 25th, the endowment of a James Watt Chair in Birmingham University, and the erection of a James Watt Memorial Building which would serve as a museum of Watt, Murdock and Boulton relics and a meeting place for engineers.

THE FULLERPHONE.

On April 24th Major A. C. Fuller read a paper before the Institution of Electrical Engineers in which he described the Fullerphone and its application to military and civil telegraphy.

During the early period of the war it was discovered, he explained, that the vibrator system of telegraphy, using the telephone receiver, which

system had shown itself admirably suited to military requirements in previous wars, possessed certain very serious disadvantages of which the following were some of the most important: (*a*) Such vibratory currents induce similar currents in other lines in their proximity; hence the number of possible lines on one route was strictly limited owing to the interference. (*b*) Owing again to the vibratory character of the current, the length of lines over which the vibrator can be used is strictly limited by their capacity. (*c*) By the use of suitable apparatus these vibrating signals can be read at immense distances from the line, and hence any messages can be overheard by listening sets. (*d*) The use of vibrators in any area effectually jams in that area friendly listening sets which might otherwise overhear enemy messages. (*e*) It is manifestly impossible to superpose the vibrator system on telephone lines.

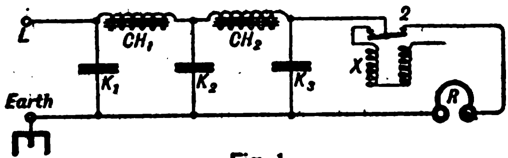


Fig. 1

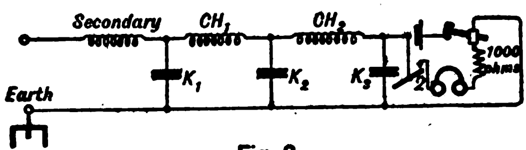


Fig. 2

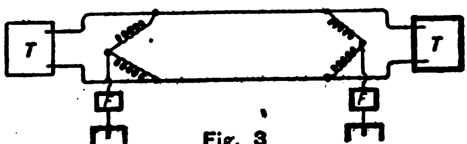


Fig. 3

"THE ENGINEER" SWAIN Sc.

In order to produce something more suitable than the vibratory system, it was necessary to retain as many of its good points, at the same time substituting something less troublesome than the vibratory current. The solution appeared to be the provision on the line of a direct current, as in ordinary single-current Morse, but of as low a value as possible—at the same time converting the energy thus transmitted into vibrating or pulsating current at the distant end—and further, such vibrating or pulsating currents had essentially to be closely confined to the distant end and prevented from surging back along the line. These ideas are embodied in the receiving and sending circuits shown in Figs. 1 and 2. K_1, K_2, K_3 are condensers, each of 1 microfarad. C H_1, C H_2 are choke coils of large impedance, each about 2.4 henries and 750 ohms resistance. The chopper is an electrically-driven vibrating interrupter fitted with two contacts, one of which is used to drive it, the other to interrupt. The

chopper or buzzer can be adjusted to any frequency between 300 and 700. Now, if a steady electromotive force is applied between the line and earth, Fig. 1, and the circuit is closed at X, a steady current will flow through the choke coils, contact 2, and the receiver. If the circuit is broken at X, the current cannot flow through the receiver, but will flow into and charge the condensers. When the circuit is again closed at X, the condensers partially discharge through the receiver R.

When the interrupter X is working we therefore get an intermittent current in the ~~telephone receiver~~, giving an audible note, while, if the choke coils and condensers are suitable, the line current alternately flows through the receiver and into the condensers and remains practically constant and continuous in the line. The result is that the dots and dashes sent by the single-current Morse key at the far end are received as short or long notes in the telephone receiver at the receiving end, while the current in the line is of much the same nature as that sent by an ordinary single-current Morse set, except that it is very much smaller, as readable signals can be obtained with about ½ micro-ampère. In practice a main battery of one dry cell is used.

The arrangement of choking coils and condensers not only prevents any appreciable variation in the line current, but also prevents any vibrating currents—such as are produced by induction from other circuits, by a buzzer on the line, or from a telephone—from passing through the operator's head telephones.

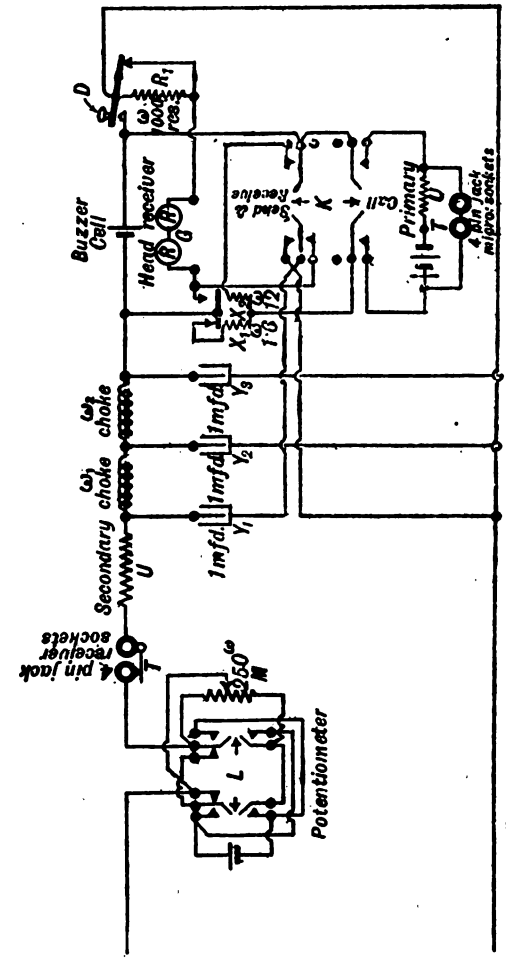
When the sending battery is connected as shown in Fig. 2, the current can only rise in the line comparatively slowly, owing to the effect of the capacity of the condensers and the self-induction of the choke coils. The object of this retardation of the rise of current is to prevent clicks being heard in a telephone receiver inserted in the line. This has a twofold object: (*a*) To prevent the possibility of Morse messages being read from these clicks; (*b*) to prevent the clicks interfering with telephony carried on over the line simultaneously with Morse signalling.

It is apparent that a telephone can be added on the same line without any interference between the Fullerphone and the telephone as far as speech is concerned. Fig. 3 shows how the Fullerphone may be—and is, in fact—superposed over telephone pairs.

The interrupter buzzer is used for calling with the Fullerphone, and for this purpose its ordinary battery is reinforced by two additional cells, making three in all. This call is considerably louder than the Fullerphone signals if the line is comparatively short and in reasonable condition, but if the line is of very high resistance or capacity the Fullerphone signals will be louder than the buzzer signals. It must be remembered that the buzzing call has all the disadvantages of the buzzer, and consequently must never be used near the front line for sending any call or signal which it is important that the enemy should not overhear.

The author then proceeded to discuss the effect of earth and leakage currents on the working of the instrument. Such disturbances may be eliminated by means of a potentiometer, which is connected in series in the line and impresses a portion of the electromotive force of the cell on the line. If the electromotive force impressed on the line is equal and opposite to that producing the earth current, the earth current and its disturbances are eliminated. Fig. 4 shows the circuits of the latest pattern Army field set. This instrument provides for telegraphy and telephony either independently or simultaneously on the one line without further apparatus. When the receiving buzzer is not working the buzzing call from a distant station is heard both in the head receivers and in the hand set, or, if the hand set is not plugged in, only in the head receivers. When "Fullerphoning" the Morse signals are heard in the head receivers only. When speaking and telegraphing the speech is heard in the hand set and the Morse signals in the head receivers.

Major Fuller than described a number of tests which were carried out before the system was adopted for the Army. Fig. 5 is a reproduction of oscillograms made by the Research Department of the G. P. O. Curves A and B are to the same scale and taken under normal conditions, and show that the intermittent currents in the receiver are considerably greater in magnitude than the line current due to their short duration. The normal variation in the line current is scarcely discernable, but this is shown in curve C by using an excess line voltage.



"THE ENGINEER" SWAIN SC

FIG. 4—ELECTRICAL CONNECTIONS OF FULLERPHONE ARMY FIELD SET

Major Fuller dealt with the possibilities of the instrument for civilian uses, and summoned up the present position as follows: For military work the Fullerphone has practically superseded the vibrating system formerly employed. It is now being tried for working long lines in back areas, where it will almost certainly replace all hand-worked Morse sounder sets. It appears, therefore, that as far as line telegraphy is considered,

the Fullerphone system will be used throughout, except—for the present, at any rate—for those lines normally working high-speed Wheatstone. For civilian use one cannot expect sounder sets that are established and giving satisfaction over well-constructed routes to be discarded in favor of Fullerphones; but it is expected that the advantages to be derived from the new system will lead to its adoption in countries not already thor-

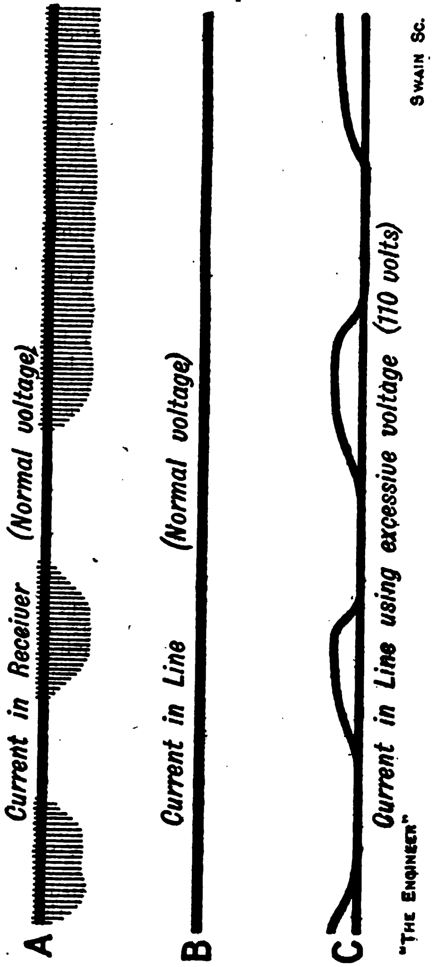


FIG. 5—OSCILLOGRAMS OF CURRENTS IN RECEIVER AND LINE CIRCUITS

oughly developed from a telegraphic standpoint. An opening should also arise in tropical countries where maintenance difficulties are considerable, and it is probably not too much to say that the system described would give satisfaction wherever difficulties from any cause render Morse sounder working precarious. For cable work the author feels that an investigation of the possibilities of a system combining Fullerphone, amplifiers, special existing relays, and recording and printing devices, would prove very profitable.—"The Engineer."

INVESTIGATIONS INTO THE CAUSES OF CORROSION OR EROSION OF PROPELLERS.*

By The Hon. Sir Charles A. Parsons, K.C.B., F.R.S., and Stanley S. Cook.

The corrosion or erosion of propellers has for many years engaged the attention of engineers and shipbuilders, and the capricious character of the action has rendered it difficult to assign an adequate and satisfactory cause to account for the observed results. At the suggestion of Professor H. C. H. Carpenter, a sub-committee of the Board of Invention and Research was formed in 1915 to investigate this subject. Those serving on this committee were Messrs. S. W. Barnaby, H. C. H. Carpenter, S. S. Cook, J. H. Gibson, Engineer Vice-Admiral Sir George Goodwin and Sir Charles A. Parsons, with Engineer-Commanders Hawkes and S. R. Dight as secretaries. The Admiralty supplied data in their possession on this subject, and information was also collected from other available sources. The investigations occupied about eighteen months.

The possible causes of corrosion or erosion considered may be stated under five headings: (1) The nature of the surface of the metal and the state of the initial stress on this surface; (2) stresses in the blades under working conditions; (3) impingement of water at high velocity on the surface of the blades; (4) cavitation; (5) water hammer produced by the closing up of vortex cavities. Each of these possible causes in turn was made the subject of investigation, and the following is a general account of the results:

(1) NATURE OF THE SURFACE OF THE METAL AND STATE OF INITIAL STRESS IN THE SURFACE.

The machining of propeller faces was reported to check erosion of the propeller blades to some extent, but examination of the material of a propeller blade before and after machining, carried out by Professor Carpenter, showed no difference in the metal except for a thin film of scale, &c., on the surface, about one-thousandth part of an inch in thickness. Nevertheless, there are advantages in romoving this scale before the propeller is put into use.

Large propeller blades which, of course, cool slowly, evidently anneal well while cooling in the mold. This has two beneficial results: (1) Casting stresses are removed from the metal; (2) inhomogeneities present in the metal as cast are obliterated.

One firm had made a practice of hammering or planishing the driving faces of blades after machining, with a view to closing up minute openings that might lead to sub-surface cavities, but the advantages, if any, are open to question. On the other hand, the locally stressed metal surface is possibly more liable to corrosion. A comprehensive examination was made by Professor Carpenter of one of the eroded blades of a fast cruiser. It failed to show any connection between the structure of the metal and the distribution of the erosion.

(2) STRESSES IN THE BLADES UNDER WORKING CONDITIONS.

The committee initiated special experiments to determine the action of sea water on manganese-bronze under stress. These experiments were carried out by Professor Carpenter at the Imperial College of Science and Technology.

* Institution of Naval Architects. Abridged.

For studying the stress below the elastic limit, two machined strips of the metal, 1⅛ inches thick, which were perfectly flat in the unstrained condition, were strained by clamping the ends firmly together, while at the middle they were held apart by a ⅜-inch rod of the same metal. The assemblage was suspended in a glass tank by glass rods and was immersed to about the middle line in sea water. The whole of the metal in this apparatus consisted of manganese-bronze of the same composition, so that all disturbing effects which might be caused by electrolytic action between different metals were eliminated. The sea water used was obtained from the Channel off Dover. The concentration of the water was kept constant and the charge in the tank was renewed every week. This experiment, made both with cast manganese-bronze and with rolled manganese-bronze, did not show in either case that a steady stress throughout the range given above caused any acceleration in the rate of corrosion of the metal in sea water.

For stresses beyond the elastic limit the test plates were removed from the apparatus and cleaned, bent into a horse-shoe form by steady squeezing in a vice, and the ends clamped in position by a steel nut and bolt insulated from the metal. They were then half-immersed in sea water, the entire gripping arrangement and the ends of the strip, for a distance of 1½ inches being covered with paraffin which protected them from contact with the sea water. The sea water was changed from time to time and the experiment continued for one month. It was found that no action took place in either test, even at the position where the tensile stress would be a maximum; in other words, that steady stress well above the elastic limit, applied both in tension and compression, does not appear to cause any acceleration of the rate of corrosion in sea water.

For alternating stresses, the test piece consisted of a strip of metal 3 inches wide, 10 inches long and ⅛ inch thick. This test piece was placed endwise in a tank of the same metal, one end being fixed to a stationary clamp attached to one end of the tank, the other receiving an oscillatory motion from a moving arm passing through the sides of the tank. The water level in the tank was so adjusted that only half the test piece was immersed, thus allowing any acceleration of the corrosion at the surface of the water to be readily observed. The speed of oscillation was kept at about sixty per minute, and the total stroke 15/16 inch, this being found to be the largest stroke which could be employed without fracture of the specimen after a few oscillations. In various tests carried out with this apparatus using specimens of both rolled manganese-bronze and cast manganese-bronze, and lasting over periods varying from 167 hours to 358 hours, one alloy commonly employed for propellers was found to be almost free from any corrosive action, whilst a second showed only a very slight increase in the rate of corrosion in consequence of the alternating stress, and quite insufficient to account for the condition of some of the propellers which were examined by the committee.

Experiments were also made at the Turbinia Works, Wallsend, which showed that the alloys commonly used for propellers when subjected to steady stress, varying from 13,500 pounds per square inch in tension to the same value in compression, showed no increase in corrosion due to stress when subjected, in addition, to the action of a diverging jet of sea water, in the apparatus described below—the specimen being a rod placed centrally in the diverging passage, whilst the initial pressure of the jet was sufficient to cause cavitation.

These experiments, therefore, gave valuable negative results, since the frequently observed bending of the tips of eroded blades had been thought to indicate that stress was an important factor. Calculations were made independently by Messrs. Barnaby, Gibson and Cook of the distribution of pressure and the working stresses on a propeller blade, with the con-

clusion that these stresses did not normally reach excessively high values. In the case, for instance, of the propellers of a fast cruiser, the maximum estimate of the stress was from 9,000 pounds to 10,000 pounds per square inch, and this only at the root of the blade and far removed from the place of corrosion. An estimate of the position of center of pressure furnished by the National Physical Laboratory from their experimental data of aerofoils did not indicate that any high stresses were likely to arise through irregular distribution of the pressure.

(3) IMPINGEMENT OF WATER AT HIGH VELOCITY ON THE SURFACE OF THE BLADE.

Several experiments were made with jets of water, of pressures up to 1,500 pounds per square inch, on plates of various brass alloys, including those used for propellers, with the water impinging both normally and obliquely. These produced merely an etching effect on the surface of the metal. It should be remarked, however, that the scale of these experiments was small.

(4) CAVITATION.

Numerous experiments were carried out at the Turbinia Works, Wallsend, with jets having diverging mouthpieces, supplied with sea water under pressure by a centrifugal pump, and discharging into a tank at atmospheric pressure. The velocity of the water was more than sufficient to sustain a vacuum and rupture of the water at the throat, in virtue of the diffuser action of the diverging mouthpieces. Under these conditions cavitation took place. A rod of the material to be tested, after being highly polished, was placed along the axis of the jet and there exposed to the action of the cavitating stream. The diagrams show the jets with the rods in the positions in which they were tested . The sea water was obtained from the North Sea, 3 to 4½ miles E.N.E. of the mouth of the river Tyne, at flowing tide.

In a second series of these experiments, carried out with the object of verifying the preliminary conclusions and of testing the effects under these same conditions of the additional circumstances of (1) stress, (2) the injection of the gases oxygen and CO_2 into the stream, and (3) annealing, a larger nozzle was employed, and a more powerful pump to give higher jet velocities. This nozzle was formed by two shaped slabs of brass fitted between parallel walls of glass. It was thus of rectangular cross section, 1 inch square at the throat, increasing to 1 inch by 4½ inches at the discharge. The rods were either ¼ inch diameter or ¼ inch square. The speed of the water was in all cases in the neighborhood of 90 feet per second, and under this condition cavitation occurred at the throat and extended for 5 inches or more beyond it.

In all, twenty-four different rods were submitted to this treatment. One of these was of brass, one of Monel metal, two of copper, fourteen of rolled manganese-bronze, and six of cast manganese-bronze—five of these were made from strips cut from propeller blades, in one case from a blade whose working face was deeply eroded.

The general character of the action on these rods, as revealed by the microscope, may be described as an etching of the surface, varying in degree. In the case of the Monel metal and the brass rods this action was very slight. In general, on the rods of manganese-bronze, there was incipient etching near the entry, increasing in depth towards the throat where the velocity was greatest. In some cases it continued deep to the end of the rods, and in other cases it gradually diminished from the throat onwards. In the incipient etching the chemical action by the salt water had just revealed the crystal structure. The rods were, to begin

with, smoothly polished, so that the action was precisely similar to the etching of a specimen by acid for examination for microstructure. In the deeper etching, however, the material between the grains, and sometimes portions of the grains themselves appeared to have been eroded away to a greater or less degree in different parts. Once the surface of the metal had been attacked the appearance remained the same, however deeply the etching penetrated, and the difference in degree was then only detected by measurement.

The general conclusion was that the degree of etching of these rods depended chiefly upon the duration of the experiment and the velocity of the water, and was not seriously influenced by liberation of gas, by stress, or by annealing; also that the effect observed did not exactly resemble either in character or in intensity the observed action on a propeller blade. Such action as was found appeared not to be attributable in any substantial degree to the cavitation, so that if, as was generally suspected, the erosion of propellers is associated with cavitation, some other and more intense form of cavitation must be sought which was not present to any great extent in these experiments.

(5) WATER HAMMER PRODUCED BY THE CLOSING UP OF VORTEX CAVITIES.

Cases of rapid and deep erosion of propeller blades and the denting of the bosses, notably on the after propellers of the *Cobra* in 1902, and on the after propellers of fast cruisers, as well as instances of undoubted deformation of metal by water hammer in water syrens, air pumps and water pumps, led the committee to desire a theoretical and practical investigation as to the possible pressures that might be produced by the closing up of vortex cavities. The calculations which were made by Mr. S. S. Cook showed that very large pressures might be generated when vortex cavities are caused suddenly to collapse by the final concentration of the initial energy upon a small volume of the fluid. The supposition is made that a certain portion of the fluid occupying, say, a sphere of a known radius is suddenly annihilated, or, what is the same thing, a rigid inner boundary to the fluid enclosing a vacuous space is suddenly removed, and the problem is to obtain the subsequent velocity with which the adjacent fluid flows into the cavity and the instantaneous pressure thus produced.

The calculations show that when a spherical vortex cavity in the sea closes upon a central nucleus of one-twentieth of the diameter of the original cavity, the instantaneous pressure or blow upon this nucleus may reach 68.2 tons per square inch, and if the diameter of the nucleus is one-hundredth of the diameter of the original cavity the pressure may reach 765 tons per square inch. It also is shown that the pressure on impact is independent of the size of the cavities, and is only a function of the ratio of the initial to the final radii, so that very small cavities may cause erosion.*

A review of the instances of propeller erosion shows them not to be inconsistent with such a theory.

In the case of the erosion on the blades of the after propellers of the *Cobra*, there was also intense water hammer on the bosses, the bronze cover plates over the set bolts attaching the propellers to the shaft being driven in with such force as to take an impression of the numbers stamped on the bolt heads. More recently, similar action was observed on the after propellers of some fast cruisers, on those parts of the propeller blades which passed through the wake of the forward propellers. The

* Mr. Cook's calculations and conclusions were confirmed in a paper to the Royal Society by Lord Rayleigh August, 1917.

forward propellers of one of these cruisers which showed no signs of erosion soon began to erode when transferred to the inner shafts, indicating that the erosion was due to the position of the propeller.

An exhaustive macroscopical and microscopical examination was made by Professor Carpenter of the blade of a cruiser's propeller which had been found to be eroded over nearly the entire surface, and which was excessively eroded in three well-defined areas. The effects of water hammer action were looked for, but very little evidence could be obtained owing to the similarity of structure of slightly eroded metal, and that subjected to hammer blows. In two places, however, evidence of somewhat excessive deformation was found which may be attributed to this cause.

An analysis made of the surface layers of the eroded portions showed no difference in composition between them and the body of the underlying metal, and therefore gave no evidence of selective corrosion. This theory of the action seemed also to be borne out by the variable positions of the patches of erosion, which were found on the blade tips, near the center of surface, at the roots, or on the boss; but always in the same places on the blades of the same propeller. The blades in their rotation apparently cut the vortices from the forward propellers, or from the shaft struts, at the same place in turn, which would account for the similarity of pattern observed. When propeller blades are found to be bent, the eating away of the surface is usually found on the side convex to the bend, and the bending might be accounted for by the water-hammer action stretching the surface on that side. Erosion is also found where there is any inequality of the blade surface, *e.g.,* at the trailing edges of shackle holes through blades, and even center-punch marks show the same effect, appearing like little comets, of which the center-punch mark forms the head.

The time element in cases of propeller deterioration is of interest. In one case, at least, the propellers had to be renewed after a three hours' trial. Apparently the action, if it is going to occur, is most severe during the early life of the propeller and proceeds more slowly afterwards. This might be accounted for by the erosion carrying the surface beyond the region of the nuclei of pressure—as in the siren experiment, described later. Rapid erosion has occurred on the casings of air pumps of the drum or Roots blower type, on the delivery side, where the vacuous spaces would close up by the ingress of the sealing water. Also in a screw pump for circulating fresh water through a condenser, where cavitation was known to exist from the usual crackling or water-hammer noises, erosion was found all round the liner and opposite to the blade tips, where cavities caused by the blades would be closed up rapidly.

Also in cases where the blades and hubs of propellers have been working in the wake of forward propellers, they have not only been rapidly eroded, but have in some cases shown the appearance of having been violently hammered, and near the tips the blades have been bodily bent.

To connect the ideal problem with the actual it remains to form a conception of the manner in which the cavities collapse in the actual case. It would seem that for our conclusions to apply we must imagine a discontinuous action, that is, that the vacuous spaces are being continually formed and collapsed, rather than that they disperse along the surface as they move away from the place of their origin. And this may be the explanation of the capriciousness of the phenomenon. In a case, for instance, such as that of the *Cobra's* propellers, where there is reason to suppose that the after propeller blades cut across vacuous spirals trailing from the forward ones, we have here the very conditions for such a discontinuous action, and the pressure on the working faces of the after propeller blades would materially assist the collapse.

CONCLUSIONS.

The conclusions arrived at are that the corrosion of propellers is very slight, but that erosion is serious and is caused by the hammer action of the water on the propeller blades, produced by cavities closing up on the surface of the blades. This action may be caused either by cavitation of the propeller itself, occurring more generally when the propeller is in a varying wake, or by the cavities and vortices formed by the action of other propellers ahead of it, and the erosive action is generally aggravated upon a propeller which works in the wake of another.

The water-hammer action is also likely to be produced when violent and abrupt eddies are formed in the water by the form of stern frame, shaft bossing or "A" brackets, or when the lines are very full and such as to cause an eddying wake. Cavitation will only produce erosion when accompanied by conditions which cause the cavities to collapse in such a way and in such a position that the energy of collapse is concentrated on a small portion of the propeller surface.

From the calculations it appears that the pressure of the water hammer is independent of the size of the cavity, depending only upon the ratio of its contraction, so that the cavities causing the erosion may be large or small. In the case of the propeller of a fast cruiser, inspected by the committee at the works of Messrs. Stone, some of the cavities which caused the erosion must have been large, as the dents observed in places were similar to those produced by using a round-nosed hammer. In the case of a certain single-screw ship of 11 knots speed, whose stern lines are full, the erosion of her propeller may have been caused by the cavities formed by the blades on their passage through the following wake, such cavities being carried with the blades and closed up suddenly when the blades entered those stream-lines of the vessel where the ship was normal, viz: about 11 per cent.

In the case of twin-screw vessels, the propeller should be placed as far as possible from the after end of the bossing-out, consistently with satisfactory mechanical conditions of the support.

In the case of four-shaft vessels, the after propellers should, in addition, be placed as far as possible abaft the leading propellers, and as far as possible clear of the race of the leading propellers.

The form of the after edges of struts and bossings-out of very fast vessels should be designed and placed in reference to the stream-lines of the vessel with great care, so as to cause a minimum of turbulent wake.—"The Engineer."

ELECTRIC DRIVE ON MERCHANT SHIPS.

COMPARISON OF ELECTRIC DRIVE WITH REDUCTION GEARING—ELECTRIC PROPULSION OF SINGLE-SCREW CARGO VESSEL PROPOSED.

By W. L. R. EMMET.*

Justification for electric drive on merchant ships is that it affords a transmission efficiency practically equal to that obtainable with gears of suitable speed reduction, and that it accomplishes this result by a simpler and more reliable means. The writer has given much thought to this problem, both with respect to geared equipments and to electric drive, has conducted many experiments, and obtained much practical experience. A year or two ago he was of the opinion that electric drive would not be commercially justified on low-speed merchant vessels on account of higher cost and a somewhat less transmission efficiency. More correct studies of

* Consulting engineer, General Electric Company, Schenectady, N. Y.

this subject and the development of improved electrical methods have shown that the difference of transmission efficiency is very slight and that the small difference of cost which will apply to a just comparison is much more than justified by many practical advantages accomplished.

Geared turbine applications to merchant ships have been of two kinds: first, single-reduction equipments with relatively low-speed turbines, and, second, double-reduction equipments with high-speed turbines. The General Electric Company was the first to produce equipments of the latter type, which afford a less efficient transmission but admit the use of a very much better and simpler turbine. By the introduction of single-reduction turbine equipments, Parsons showed by comparative tests a reduction in steam consumption of about 19 per cent as compared with the reciprocating engine. Four years ago I predicted a further reduction of 14 per cent by the use of high-speed turbines with double-reduction gearing, and very complete data and tests have proved the correctness of this prediction.

SINGLE-REDUCTION GEARS ABANDONED IN BRITISH SHIPS.

An interesting fact in this connection is that in recent designs Parsons has abandoned the single-reduction method, and the new standard English ships are being built with double-reduction gearing and high-speed turbines. At the same time that this change is being carried on on a large scale in England, the DeLaval Company and others in this country, with the approval of the Emergency Fleet Corporation, have made the opposite change and have abandoned double reduction in favor of single reduction, on the ground that the gearing could be made more reliable. An interesting fact in this connection is that the principal troubles with marine gears have developed in the low-speed element.

These facts illustrate the uncertainties of the gear situation and the unformed state of opinion relating to it.

Machinery which drives a propeller may be subject to very severe shocks and periodic vibrations, and gear troubles have developed on ships in a manner which seems erratic and very difficult of classification.

The efficiency of ship gears has also been overestimated. Careful tests have indicated that the efficiency of transmission in a 2,500-horsepower double-reduction equipment, running smoothly and in good condition, does not exceed 94.5 per cent, and it must be remembered that the equipment is further handicapped by the presence of a reversing turbine which occasions about eight times as much friction resistance as a similar turbine moving in the ahead direction. With high-speed turbines and the most improved electrical apparatus, we can get a transmission efficiency of 93.5 per cent on such a ship, so that it can be safely predicted that in any vessel in which high-speed turbines are used, the difference of transmission efficiency will not exceed one per cent. If, on account of gearing conditions, the turbine has to be divided into two or more parts, there is a further disadvantage through increased packing losses and other complications.

EFFECTS OF TEMPERATURE WITH USE OF REVERSING TURBINES.

Another matter which is important in the comparison of geared and electric ship equipments is the effects of temperature which may arise in connection with the use of reversing turbines, particularly where high degrees of superheat are applied. In an electrically-driven ship the turbine always moves in the same direction and never requires to be stopped except when operation is entirely discontinued. Thus its service is equivalent to that of a turbine on shore. When a turbine is operated in the

reverse direction the friction is at least eight times as great as in the normal direction. The reversing elements of our 2,500-horsepower ship turbines, although very small in diameter and of a minimum rotation loss, will, when operated in a 10-inch vacuum at full speed, heat up enough to turn blue. In our tests we regularly avoid this by introducing water. The introduction of water or any fluid which may limit this heat will naturally cause a large increase of loss.

An experiment which suggests the possibilities of such heat troubles as have been mentioned was recently made in Schenectady. A turbine designed to operate at 3,600 revolutions per minute was forced to revolve in the reverse direction at 2,000 revolutions per minute. A small amount of steam, passed through the buckets in the normal direction while it was being reversed, produced in a short time a temperature of 940 degrees F. in the nozzles of the last stage.

Temperature variations in actual practice in ship turbines are, to a great extent, limited by the heat-storage capacity of the turbine parts themselves, but there are bound to be local heatings and irregular distributions of heat, which, with high superheat, cannot fail to be a source of increased danger. Superheat affords means of greatly increased economy wherever turbines are used, and this advantage should not be sacrificed in ships where fuel economy is of such great importance. In an electrically-driven ship superheat can be a source of no disadvantage to a turbine—in fact, it is likely to increase the life of the blading.

ELECTRICAL PROPULSION LIMITED BY PROPELLER SPEED.

The electrical propulsion of ships may be limited under some conditions by the matter of propeller speed, since we cannot have less than two poles in the generator, and since it is undesirable to have to provide more than 60 or 70 poles in a motor. For ships requiring 3,000 horsepower or over and a speed of about 11 knots, a turbine operating at 3,000 revolutions per minute can be advantageously used, which, with a 60-pole motor, will give 100 revolutions per minute to the propeller, and a high propeller efficiency. In higher-speed vessels the conditions are generally easier. With smaller powers and propellers operating under 100 revolutions per minute it will generally be desirable to use gears, even if electrical transmission is adopted. In that case it is desirable to place the gears between the turbine and the generator instead of between the motors and propellers, as has been done.

DIFFICULTIES IN LOW-SPEED ELEMENTS OF GEARING.

Experience with ship gearing indicates that the greatest difficulties arise in the low-speed elements. This is probably due to the fact that these elements receive more direct shocks from the propeller, and also that they give a much less perfect lubrication effect. There is presumably an effective oil cushioning in high-speed gears which is absent in low-speed gears.

In electric drive equipments it would generally be desirable to furnish direct current exciting current from a separate source, and it has been thought desirable to combine excitation, lighting and the driving of auxiliaries. The auxiliaries on most existing ships involve a considerable unnecessary loss of power, and by driving them electrically and exhausting a well-designed auxiliary generating unit into the feed heater and into an intermediate stage of the turbine, a very good efficiency and simple operating condition can be afforded.

The following copy of a proposal recently made illustrates the possibilities of an application to a 4,000-horsepower vessel with a single screw operating at 110 revolutions per minute:

PROPOSAL FOR ELECTRIC DRIVE ON 4,000-HORSEPOWER MERCHANT VESSEL.

"The auxiliary generating units are of such a size that one of them will be capable of affording lighting and excitation and all the auxiliary power which has been contemplated while the ship is at sea. These units are designed to run condensing or non-condensing with a good water rate. By carrying as a spare part a suitable direct-current motor and gear which could be connected to the end of the main shaft, these auxiliary units are used to propel the ship at nearly half speed in case the main unit or main motor were unfit for service. It would in any case be desirable to have auxiliary condensing facilities for use in port, and these could be used in such an emergency.

"The main propulsion equipment consists of:

	Approximate Net Weight, Pounds
One 3,200-k. v. a., 3,500 revolutions per minute, 2,500-volt turbine generating unit	82,000
One 4,000-horsepower, 110 revolutions per minute, alternating-current motor	80,000
Instruments and controlling mechanism for the above	9,000

"Our calculations concerning the performance of this equipment show the following results, and, since the efficiency of every part is thoroughly understood, it is certain that these results can be actually obtained in practice. When the turbine is operating with a steam pressure of 265 pounds gage at the throttle, 150 degrees F. superheat, and 28.5 inches vacuum, we should obtain a brake horsepower delivered to the propeller shaft with 9.4 pounds of steam, not including auxiliary steam.

"In giving guarantees, we must make an allowance for possible errors in testing and will guarantee that this water rate, under these conditions, will not exceed 10 pounds per shaft horsepower hour.

"*Auxiliaries.*—We have assumed that the boiler-feed pump is steam-driven and requires 2,500 pounds of steam per hour, also that a Blake twin-beam wet and dry-air pump is used which consumes 600 pounds of steam, making a total of 3,100 pounds of steam from these pumps for feed-water heating. The following electrically-driven auxiliaries have been assumed:

	Motor speed.	Horsepower.	Electrical Input, Horsepower.
Circulating pump	1,700/1,000	75.00	82.5
Excitation		80.00	80.0
Lighting		20.00	20.0
Oil Cooler	1,700	1.25	1.5
Fire room blowers	850	20.00	22.5
Steering	800	5.35	6.0
Continuous load			212.5 = 157.0 kw.

	Motor speed.	Maximum Horsepower.	Average Continuous Horsepower.
Sanitary pump	1,700	5.0	2.5
Fresh water pump	1,700	5.0	2.5
Refrigerating pump	1,700	5.0	2.5

Evaporator pump	1,700	1.0	1.0
Workshop	1,700	5.0	2.5
Bilge pump	850	5.0	5.0
			16.0
Total input			18.0 = 13.5 kw.
Total maximum load			170.5 kw.

"In addition to the above motors there is included one 20-horsepower, 900 revolutions per minute induction motor for operating the blower for main motor ventilation. In connection with the above motors, while we are including hand-operated starting devices, no mechanical parts such as pumps, blowers, etc., are included. In other words, only motors and starters are included.

"We propose to drive these auxiliaries from two 150-kilowatt, direct-current sets, operating non-condensing against a back pressure of 5 pounds gage, with part of this exhaust used to bring up the feed-water temperature to 212 degrees F. and the remainder going into a suitable stage in our turbine for additional work.

"The total flow of steam from the boiler at maximum speed will be 43,000 pounds per hour. Assuming 14 pounds of steam evaporated per pound of oil (same efficiency as Babcock & Wilcox boilers on battleship *Wyoming*), this would require 3,100 pounds of fuel oil per hour, or 33.25 tons (2,240 pounds) for 24 hours. Thus the water rate, including all auxiliaries, lighting, and steam required for heating feed water, should be 11.1 pounds per shaft horsepower hour.

"The electrical apparatus proposed for this purpose is of an extremely simple and reliable type. Our records over long periods of years covering such apparatus show that only about one-tenth of one per cent of the motors and generators of such voltage, which are installed in all kinds of service, show any electrical trouble in a period of ten years.

"The turbine is of the most efficient type manufactured for such a capacity and is designed with very liberal factors of safety with all the most recent features of construction. Such a turbine is capable of running at a considerable proportion of its capacity if one of its many wheels is in operating condition. Packings, bearing sleeves and other perishable parts are easily replaceable, so that under almost any combination of circumstances such a turbine can be quickly got into operative condition after it has been accidentally damaged.

"The standards of reliability are so high with such apparatus that it is believed that they occasion less risk of stoppage than is incurred in single-screw ships propelled in any other manner. In case, however, the novelty of such an equipment raises doubts which may increase insurance rates, a motor can be provided as stated above which will propel the ship at reduced speeds from power delivered by the auxiliary generating units.

"It is believed that the operation of these relatively large auxiliary generating units and the use of motors for driving the auxiliaries mentioned are justified from the standpoint of economy. Generating units of such a capacity will give a very good efficiency, and their operation involves no more complication of detail than is occasioned by the use of the smallest steam turbine—in fact, the details of the larger turbine are simpler and better than those of the smaller one."

The proposal quoted above describes an equipment which is in itself much less liable to trouble or interruption than any type of existing equipment used on a single-screw ship, and at the same time affords, at very

small cost for spares, means by which the ship can be run if the main machinery is inoperative. The figures given as to results are correct and dependable, and are so good that their accomplishment in many classes of service would pay for the change in three years. With very few exceptions, the prevailing opinion of the most experienced marine engineers is that the reciprocating engine for ship propulsion is obsolete. Its replacement is inevitable, and the General Electric Company, with its experience and facilities for development, is ready to do the work.—"Marine Engineering."

OBITUARY.

CHIEF ENGINEER CHARLES H. MANNING, U. S. N.

Chief Engineer Charles H. Manning died at his home, in Manchester, N. H., on the 8th of April, 1919.

He was born in Maryland in 1842, where his people had lived for more than a century. He was graduated at Harvard University, in the class with John D. Long (afterwards Secretary of the Navy), and Eugene Hale (so long a U. S. Senator).

He entered the Navy as a Third Assistant Engineer in February, 1863, and was assigned to special duty in Baltimore. It was an unfortunate order for him, as it delayed his promotion, having deprived him of the essential sea service. In 1865 he was assigned to the *Dacotah,* on which vessel he served three years, a period necessary for his promotion to Second Assistant Engineer. His next cruise was on board the *Seminole,* on the Home Squadron. He was promoted to Passed Assistant Engineer in 1872, and ordered to the Naval Academy as instructor.

In 1875 he was ordered to the *Swatara,* the first ship of the Navy to make a voyage with a compound engine. The voyage was to the Kerguelen Islands, in the South Sea, for the purpose of the observation of the transit of Venus, and was a hazardous voyage, but productive of grand results.

His next duty was at the Naval Academy, where he remained three years as instructor. He served as Chief Engineer on board the *Dispatch* 1881 and 1882, after which he was granted leave of absence, which was extended from time to time. During this period he was under treatment for deafness. The Navy Department favored Chief Engineer Manning, as his services were so valuable and it was desired to retain him on the active list.

He was retired in June, 1884, and at once was made Chief Engineer of the famous Amoskeag Manufacturing Establishment at Manchester, at a salary more than three times what he had received on the active list of the Navy. He was promoted to the rank of Chief Engineer in the Navy (on the retired list), with the rank of Lieutenant Commander, by act of Congress approved March 4, 1911.

The many improvements he made at the Amoskeag works, his success as a hydraulic engineer and his willingness to lend a helping hand to all his friends, soon brought him in the limelight as a Consulting Engineer.

Mr. Manning retired from business about four years ago, having become so deaf that he felt it unjust to his employers to continue. Personally he was one of the most popular men in his corps and one of the brightest and best informed. He was a member of this Society since its organization. He was a member of the Society of Mechanical Engineers, Naval Architects and Marine Engineers, and took part in their debates.—G. W. B.

MARSHALL T. DAVIDSON.

The death of Marshall Ten-Broeck Davidson occurred at his residence in Brooklyn on the 10th of April, after a long illness. He was born in Albany in 1837, but his early life was spent in Hudson. His paternal ancestry was Scotch, dating from the colonial period, and his maternal ancestry was Holland-Dutch, dating from the time of the Dutch occupation of New Amsterdam, afterwards New York City. His intimate acquaintance with the Engineers of the Navy and his interest in the profession of Steam Engineering began in his boyhood.

He was educated in the public schools of Hudson and in the Hudson Academy, and he learned the trade of machinist in New York City. On reaching his majority he went to California as an Engineer on board the steam ship *Hudson*. He

remained in this service, which traded in Puget Sound, the Columbia River and Bay of San Francisco, until the Civil War broke out, when he was employed to superintend the building of two army transports, the *Arizona* and *Clinton,* which had been taken over from the Harris and Morgan Company. When the *Arizona* was completed he became her Chief Engineer, and carried troops to the Gulf of Mexico for General Banks' Expedition to the Teche Country, where the writer first met him. He later became a Chief Engineer in the Revenue Cutter Service and later became the President of the Davidson Steam Pump Company in Brooklyn, which position he held until his death.

The Davidson Pump soon became famous, and though its price was greater than its competitors, there was sale for all that could be built. His high-duty engines for city water works, some of a capacity of 40,000,000 gallons a day, were famous.

He married Miss Bame, of Hudson, who died in 1881, leaving two daughters, one of whom (the Countess Seckendorff) survives. There are five grandchildren.

He was a member of The Union League, Hanover, Brooklyn and Manhattan Clubs, the American Society of Navy Engineers, the American Society of Mechanical Engineers, the Society of Naval Architects and Marine Engineers, and The Grand Army of the Republic.

Personally he was charming and magnetic, and had hosts of friends. He was always delighted to do a good and generous act.—G. W. B.

BOOK REVIEW.

"Wrinkles in Practical Navigation," by T. S. Lecky, Eighteenth Edition, revised and enlarged by William Allingham, reprinted by arrangement with the English publishers, D. Van Nostrand Co., $5.00 net.

Lecky's "Wrinkles" needs no introduction. First published in 1881, successive editions have been deservedly popular. The method of treatment employed is in sharp contrast to that of the "American Practical Navigator" and Muir's "Navigation and Compass Deviations," but the "wrinkles" and short cuts disclosed make the book worth the purchase price.

In this edition the greater part of the book is about the same as previous editions. One new chapter, "New Meteorological Measures for Old," discusses at some length the proposed adoption of the metric system and centesimal circular and angular units. Four new appendices are added.

It is unfortunate that a book as widely known as Lecky's "Wrinkles" should, in a 1919 edition, fail to enlarge upon the "computed altitude" methods of obtaining a line of position. The Marc St. Hilaire method, using either a sine-cosine or haversine formula or the Aquino tables, has eliminated much of the drudgery of working up sights, with the added advantage of obtaining a *good* line of position for any azimuth of the observed body.—M. W. B.

ASSOCIATION NOTES.

The following members and associates have joined the Society since the publication of the last number of the JOURNAL:

MEMBERS.

Barr, William M., 116 W. 39th St., New York City.
Brown, Ray E., Lieut. (CC), U. S. N.
Fallon, H. N., Lieut., U. S. N.
Holden, C. F., Lieut., U. S. N.
Rasch, F. W., Lieut., U. S. N.

ASSOCIATES.

Boyd, W. I., Lieut. Commander, R. F.
Price, Joseph, 203 West 11th St., New York City.
Robinson, R. L., 209 Addison Avenue, Palo Alto, Calif.
Thomas, George S., Wm. Cramp & Sons Co., Philadelphia, Pa.
Wardrop, G. Douglas, Managing Editor, Aerial Age, 280 Broadway, New York City.

Please do not fail to keep the Secretary-Treasurer advised of your address.

REPORT ON THE AMERICAN SOCIETY OF NAVAL ENGINEERS FOR THE YEAR ENDED DECEMBER 31, 1918.

Date: March 24, 1919.

THE AMERICAN AUDIT COMPANY.

F. W. LAFRENTZ, C. P. A. (N. Y.), *President.*
THEO. COCHEU, JR., C. P. A. (N. Y.), *Vice-President.* A. F. LAFRENTZ, *Sec'y and Treas*
C. R. CRANMER, *Resident Manager.*

TELEPHONE: 2705 MAIN.
CABLE: AMDIT, NEW YORK.

COLORADO BUILDING, WASHINGTON, D. C., March 24, 1919.

THE COUNCIL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS,
Washington, D. C.

Dear Sirs: We have audited the books and accounts of THE AMERICAN SOCIETY OF NAVAL ENGINEERS for the year ended December 31, 1918, and submit our report, including Exhibits, as follows:

Exhibit "A."—Balance Sheet as at December 31, 1918.
"B."—Statement of Income and Expenditure for the year ended December 31, 1918.

The result of the operation of the Society for the year shows a net profit of $4,269.71.

We checked the cash to March 20, 1919, at which date the Cash Book called for $8,056.57, of which $7,858.57 was on deposit with the American Security and Trust Company; the balance, $198.00, represented receipts not deposited.

We verified the securities of the Society by actual inspection.

Our examination of the financial operations of the Society, together with the assurances of the Secretary, shows that the Society was on a cash basis at December 31, 1918, with no liabilities.

Respectfully submitted,

THE AMERICAN AUDIT COMPANY,
By C. R. CRANMER, *Resident Manager.*

Approved:
F. W. LAFRENTZ,
President.

[SEAL.]

Attest:
C. W. GOETCHINS,
Ass't Secretary.

EXHIBIT "A."

AMERICAN SOCIETY OF NAVAL ENGINEERS,

Washington, District of Columbia.

Balance Sheet, As at December 31, 1918.

ASSETS.

Cash in bank	$3,527.09	
Certificate of deposit, Munsey Trust Company	1,000.00	
		$4,527.09
Account Receivable:		
Dues	$1,496.25	
Publication sales	111.24	
Subscriptions	2,062.14	
Advertisements	1,708.00	
		5,377.63
Investments:		
Army and Navy Club bonds	$1,000.00	
Liberty Bonds, First Issue	1,000.00	
Liberty Bonds, Third Issue	1,000.00	
Liberty Bonds, Fourth Issue	500.00	
Washington Railway & Electric Co. bonds	4,320.00	
War Saving Stamps	838.00	
		8,658.00
Furniture and fixtures		50.84
Gold medal		16.00
		$18,629.56

LIABILITIES.

Surplus:		
Balance, January 1, 1918	$14,359.85	
Net profit for the year ended December 31, 1918, Exhibit "B"	4,269.71	
		$18,629.56
		$18,629.56

EXHIBIT "B."

AMERICAN SOCIETY OF NAVAL ENGINEERS,

Washington, D. C.

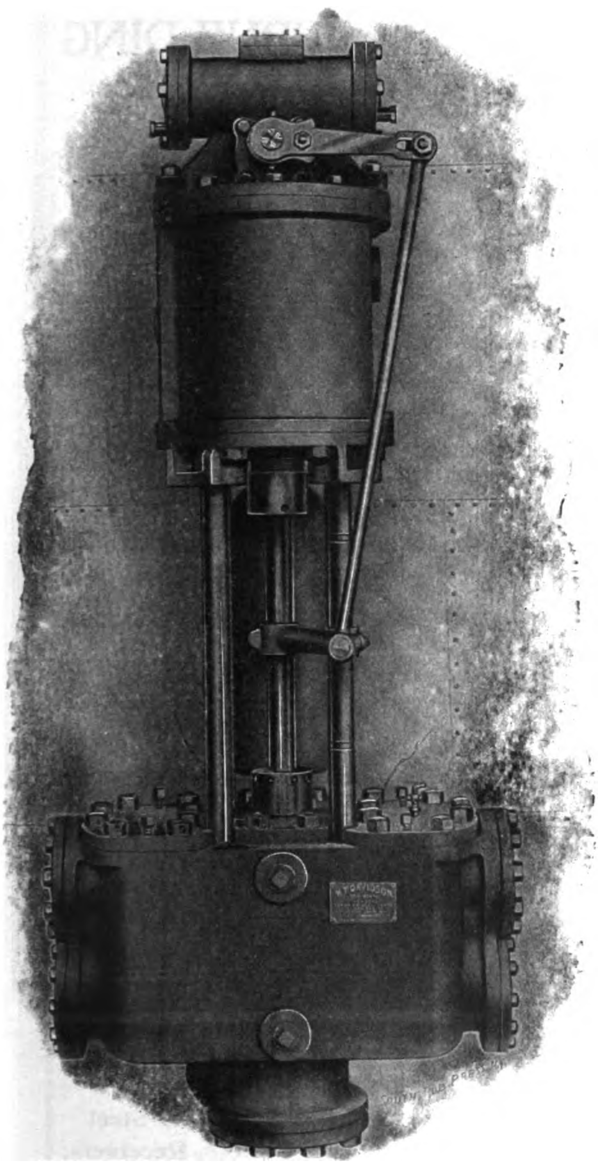
Statement of Income and Expenditure, for the year ended December 31, 1918.

INCOME.

Dues		$5,846.66
Publication:		
Subscriptions	$4,221.90	
Sales	585.79	
Advertisements	2,363.50	
Exchange	2.89	
		7,174.08
Old dues collected		102.45
Interest on investments		472.16
		$13,595.35

EXPENDITURE.

Publication:		
Printing	$5,445.20	
Engraving	670.43	
Drafting	20.00	
Manuscript	883.50	
Postage and expressage	220.34	
Commissions	156.92	
		$7,396.39
General Expense:		
Salaries	$1,620.00	
Stationery and printing	111.85	
Certificates	17.40	
Rent, safe deposit box	5.00	
		1,754.25
Dues charged off		175.00
Net Profit for year ended December 31, 1918:		
Carried to Exhibit "A"		4,269.71
		$13,595.35



AUXILIARY MACHINERY

WITH AN ENVIABLE REPUTATION

STEERERS, HOISTERS, WINCHES
CAPSTANS AND WINDLASSES
HAND, STEAM AND ELECTRIC

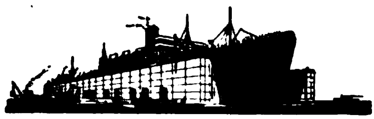
AMERICAN ENGINEERING COMPANY

PHILADELPHIA, PA.

MACHINISTS AND FOUNDERS

Successor to American Ship Windlass Co. and Williamson Bros. Co.

15-318



20,000 TONS LIFTING CAPACITY

New Floating Dry Dock at Sparrows Point Plant, Sparrows Point, Md., Now in Operation.

OTHER DOCKING FACILITIES:

PACIFIC COAST,

UNION PLANT, SAN FRANCISCO, CALIF.

GRAVING DOCKS: No. 1. Length 1096 ft. width 120 ft., depth over sill 40 ft. (One of the largest docks in the U. S.)
No. 2. Length 750 ft. width 103 ft., depth over sill 28 ft.

FLOATING DOCKS: No. 1. Length 450 ft., width 100 ft.,
No. 2. Length 301 ft., width 68 ft.,
No. 3. Length 271 ft., width 62 ft., } Maximum Lifting Capacity 10,000 tons.

MARINE RAILWAYS: No. 1. Length 320 ft., capacity 4000 tons.
No. 2. Length 220 ft., capacity 2000 tons.

ATLANTIC COAST,

HARLAN PLANT, WILMINGTON, DEL.

GRAVING DOCKS: No. 1. Length 330 ft., width 45 ft., depth over sill 13 ft.

PLANTS: Fore River, Quincy, Mass. Harlan Plant, Wilmington, Del.
Moore Plant, Elizabeth, N. J. Sparrows Point Plant, Sparrows Point, Md.
Union Plant, San Francisco, Calif.

Shipbuilders and Engineers. Marine and Stationary Engineers.
Curtis Marine Turbines. Auxiliary Machinery.
Water Tube and Scotch Boilers. Special Facilities for Repair Work.

BETHLEHEM SHIPBUILDING CORPORATION, LTD.

BETHLEHEM, PENNA.

BRANCH OFFICES: 111 BROADWAY, N. Y. 260 CALIFORNIA AVENUE, SAN FRANCISCO.

IN THE LATEST IMPROVED PATENT

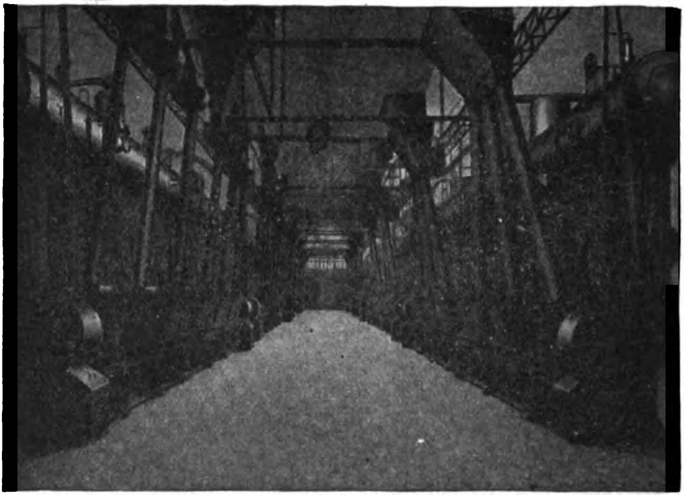
NICLAUSSE

WATER-TUBE BOILER

Incrustation is Impossible in the Tubes Exposed to Radiant Heat

These tubes being fed only with water previously automatically softened and heated at a high temperature.

3 MILLION H.P. IN USE



3 MILLION H.P. IN USE

View of 27,000 H.P. Niclausse Boiler Plant fitted with Niclausse Automatic Stokers.

NICLAUSSE
AUTOMATIC STOKERS

burn the LOWEST grades of fuel and give the HIGHEST economical results WITHOUT SMOKE.

Works and Head Office: 24 RUE DES ARDENNES, PARIS (XIX^{e}. Arr^{t}).

Telegraphic Address: GÉNÉRATEUR-PARIS Code Used: A B C, 5th Edition

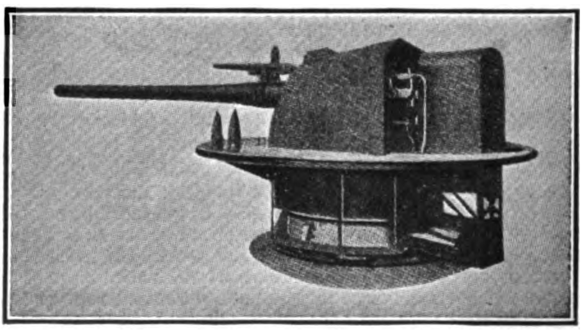
Licensees for Great Britain and its Colonies { **The British Niclausse Boiler Co., Ltd. Caxton House, Westminster, S. W.**

Telephone, 4086 Victoria Telegrams, Britniclos, London

BETHLEHEM STEEL CO.

SOUTH BETHLEHEM, PA. **25 Victoria Street, LONDON**

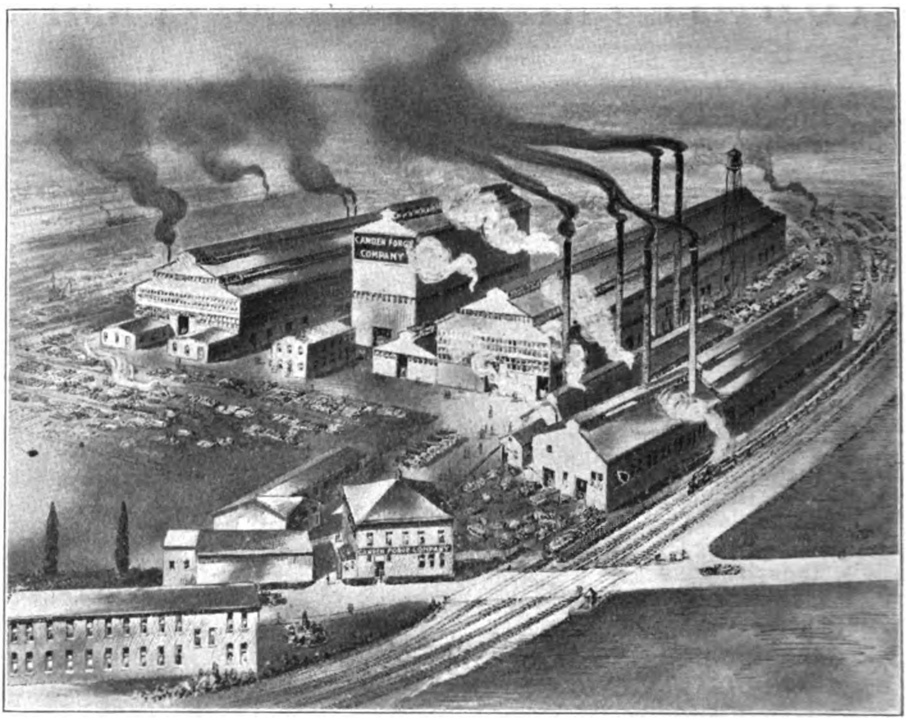
Naval, Field and Coast Defense GUNS and MOUNTS
Armor
Turrets
Projectiles



Forgings
Castings
Shafting
Rails and Structural Steel

9."2 Coast Defense Gun with Sub-calibre Equipment, on Barbette Mount.

We are continuously manufacturing ORDNANCE MATERIAL for the U. S. Army, U. S. Navy and Foreign Governments



SEATTLE CONSTRUCTION & DRY DOCK COMPANY

Seattle, Washington

**SHIPBUILDERS——ENGINEERS——
BOILERMAKERS AND FOUNDERS**

LARGEST FLOATING DRY DOCK EQUIPMENT ON THE PACIFIC COAST

COMPLETELY EQUIPPED PLANT FOR QUICK REPAIRS

Cable Address: "Threedocks"

LALOR Automatic STOP-VALVE

FOR

FUEL OIL SYSTEMS

The Valve that makes the use of Fuel Oil as safe as coal. A Perpetual Fire Life and Accident Insurance

SEND FOR BULLETIN AND REFERENCES
ON APPROVED LIST ISSUED BY UNDERWRITERS' LABORATORIES, AND USED BY UNITED STATES NAVY AND MERCHANT MARINE

LALOR FUEL OIL SYSTEM CO.,

527–529 Colvin St., Baltimore, Md.

METALLIC PACKING for the piston rods and valve stems of main and auxiliary engines, saturated or superheated steam. Entirely flexible, therefore no possibility of heating rods. In use on many ships in the Navy.

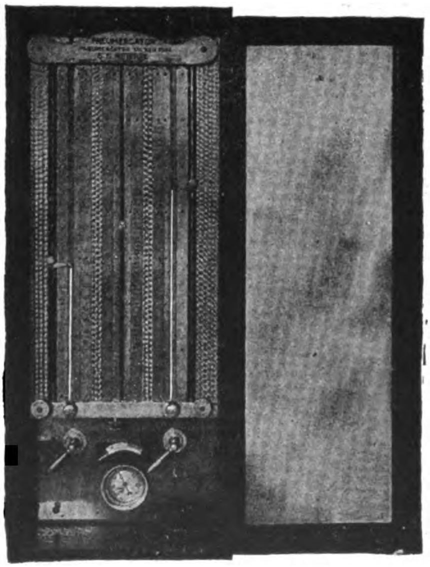
**THE UNITED STATES
METALLIC PACKING CO.
PHILADELPHIA.PA..U.S.A.**

PNEUMERCATOR

MODEL D-1—Indicates forward and aft drafts. Registers mean draft and corresponding tons displaced. Shows trim, checks invoices and deliveries, weighs cargoes and bunkers. Other types of this model indicate forward, aft and midship drafts only.

MODEL T-1—Indicates depth and volume or weight of contents of fuel oil, fresh water, ballast tanks or bilges. Checks invoices or withdrawals. Furnishes a perpetual inventory. Other types of this model can be arranged to ring a high or low alarm or both, or to start and stop an electrically driven pump.

Model T-1



Model D-1

PARTIAL LIST OF USERS

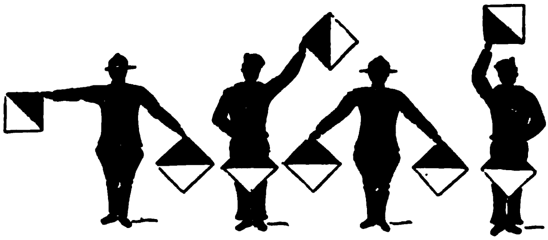
The Union Sulphur Co.
New England Coal & Coke Co.
W. R. Grace & Company
Pioneer Steamship Company
Oceanic Steamship Company
Florida East Coast Railway Co.
Mexican Petroleum Company
Warner-Quinlan Asphalt Co.
Cox & Stevens
Stone & Webster Const. Co.
General Fire Extinguisher Co.
United States Government

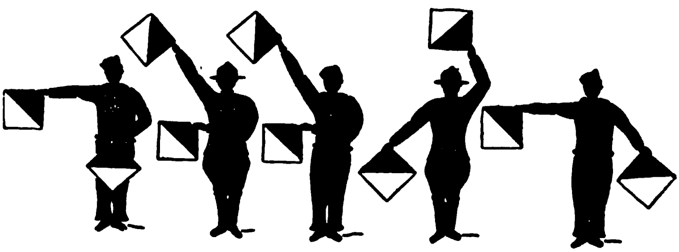
PNEUMERCATOR COMPANY, INC.

15 PARK ROW
PARK ROW BUILDING
NEW YORK, N. Y.

Harry S. Parks	H. H. Erickson	J. C. H. Ferguson	W. S. DePerris Co.
450 N. Tenth St.	208 N. 5th Ave.	Monadnock Building	103 Grand Trunk Dock
Philadelphia, Pa.	Chicago, Ill.	San Francisco, Cal.	Seattle, Wash.

F. S. Dudgeon, Ltd., 30 Great St. Helens, London, E. C.





VOL. XXXI. NO. 3.

JOURNAL
OF THE
AMERICAN SOCIETY
OF
NAVAL ENGINEERS.

PUBLISHED QUARTERLY BY THE SOCIETY, FOR THE ADVANCEMENT OF NAVAL ENGINEERING.

AUGUST, 1919.

WASHINGTON, D. C.

R. BERESFORD, PRINTER, 606 F STREET, N. W.

Entered at the Post Office at Washington, D. C., as Second Class Matter.—Acceptance for mailing at special rate of postage provided for in Sec. 1103, Act. of Oct. 3, 1917, authorized on July 3, 1918.

OFFICERS OF THE SOCIETY.

President:

Rear Admiral H. P. NORTON, U. S. Navy.

Secretary-Treasurer:

Lieutenant Commander F. W. STERLING, U. S. Navy (Retired).

Council:

Rear Admiral H. P. NORTON, U. S. Navy.
Rear Admiral T. W. KINKAID, U. S. Navy.
Captain ROBERT STOCKER, (C. C.) U. S. Navy.
Commander S. M. ROBINSON, U. S. Navy.
Commander W. W. WHITE, U. S. Navy (Retired).
Captain of Engineers M. W. TORBET, C. G.
Lieutenant Commander F. W. STERLING, U. S. Navy (Retired).

SUBSCRIPTION RATES.

The subscription price of the JOURNAL, postpaid to the United States and possessions, Canada, Cuba and Mexico, is $5.00. Single copies, $1.30. To other countries in the Postal Union, $5.40. Single copies $1.35. All subscriptions are payable in advance.

Make checks, drafts and postal orders payable to

AMERICAN SOCIETY OF NAVAL ENGINEERS.

Advertising rates will be furnished on application.

It is earnestly requested that prompt information be given of changes in address, or of failure to receive the JOURNAL.

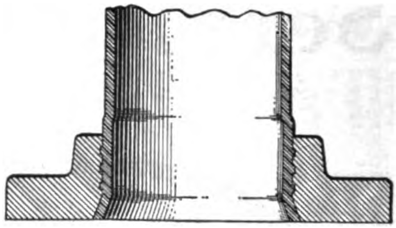
Address all communications for the Society to

AMERICAN SOCIETY OF NAVAL ENGINEERS,
Navy Department, Washington, D. C.

CONTENTS.

INDEX TO ADVERTISERS.

Name.	Manufacturers of or dealers in	Post office address.	Page.
J			
Johns-Manville Co., H. W.	Packings, Bitumen-Mastic Solution, Enamel and Cement, Audiffren-Singrun Refrigerating Machines, Navy Brand Fire Felt, "Noark" Enclosed Fuses, Pipe Coverings — Magnesia, and Asbestos.	296 Madison ave., New York City.	xxxv
K			
Kinney Manufacturing Co.	Kinney High Pressure Pumps	Boston, Mass.	xxxvii
Kelvin & Wilfrid O. White Co.	Adjusters of Ships' Compasses. Contractors for Navigational Equipment.	112 State st., Boston, Mass.	xxxi
Kingsbury, Albert, Engineer.	Thrust Bearings	Oliver Bldg., Pittsburg, Pa.	xxxii
L			
Lalor Fuel Oil System Co.	Automatic Stop-Valves	527-529 Colvin st., Baltimore, Md.	xxxii
Lidgerwood Mfg. Co.	Hoisting Engines, Electric Hoists, Conveying Machinery.	96 Liberty st., New York.	xxvi
Lovekin, Luther D.	The Lovekin Improved Automatic Assistant Cylinder for Valve Gears.	Mill Creek Road, Ardmore, Pa.	xxv
Lovekin Pipe Expanding and Flanging Machine Co.	Pipe Expanding and Flanging Machine.	421 Chestnut st., Philadelphia, Pa.	vi
M			
Mason Regulator Co.	Mason Reducing Valves and Pressure Regulators.	Boston, Mass.	xlii
Metal & Thermit Corporation.	Thermit Quarterly	Equitable Building, 120 Broadway, New York City.	xi
N			
Newport News Shipbuilding and Dry Dock Co.	Ships and Engines	Newport News Shipbuilding and Dry Dock Co., Newport News, Va.	xviii
Niclausse Boiler Co.	Niclausse Water-Tube Boilers, Niclausse Automatic Stokers.	24 Rue des Ardennes, Paris, France.	xxiv
P			
Parsons Marine Steam Turbine Co., Ltd.	Marine Steam Turbines	2 Rector st., New York City.	xiii
Phosphor-Bronze Smelting Co.	Phosphor-Bronze and Delta Metal Castings, Forgings, etc.	2200 Washington ave., Philadelphia, Pa.	xi
Pneumercator Company, Inc. Kelvin, Bottomley & Baird, Ltd.	Pneumercators	15 Park Row, Park Row Building, New York, N. Y. Glasgow, Scotland.	xl
Power Specialty Co.	Builders of the Famous Foster Superheater. Supply Duval Metallic Packing.	111 Broadway, New York.	iv
R			
Richardson — Phenix Co.	Lubrication Engineers and Manufacturers.	114 Reservoir ave., Milwaukee, Wis.	xliv
Roelker, H. B.	Allen Dense-Air Ice Machines, Screw Propellers. Consulting Engineer.	41 Maiden Lane, New York.	iv
Row & Davis, Engineers, Inc.	Paracoil Feed Water Heaters and Evaporators.	90 West st., New York.	xxiii
S			
Schmidt Co., F. J.	Naval and Army Outfitters	63 Maryland ave., Annapolis, Md.	xlii
Schutte & Koerting Co.	Fuel Oil Burning Systems, Film Water Heaters, Ship Maneuvering Valves, Ship Emergency Valves, Marine Free Exhaust Valves, Reducing Valves, Distillers, Evaporators.	12th and Thompson sts., Philadelphia, Pa.	xxix
Seattle Construction and Dry Dock Co.	Shipbuilders, Engineers, Boiler Makers, Founders.	Seattle, Wash.	xxxiii
Sterling Engine Company	Sterling Engines	1264 Niagara st., Buffalo, N. Y.	xxxix



THE LOVEKIN FLANGE

ADOPTED BY THE NAVY DEPARTMENT AND BEING USED IN OVER 6,000,000 TONS OF SHIPS NOW UNDER CONSTRUCTION.

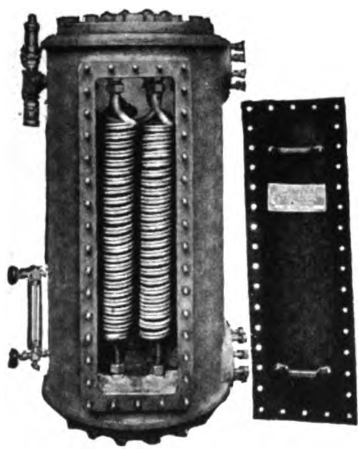
OUR MACHINES ARE INSTALLED IN EVERY INDUSTRIAL NAVY YARD, AND ARE IN OPERATION IN OR ON CONTRACT FOR THE FOLLOWING PLANTS:

Atlantic Corporation
American International Shipbuilding Corp.
American Shipbuilding Company (5 plants)
Bethlehem Shipbuilding Corporation—
    Fore River Plant
    Maryland Shipbuilding Plant
    Union Plant
    Liberty Plant
Chester Shipbuilding Company
Carolina Shipbuilding Corporation
Dierks-Blodgett Shipbuilding Company
Downey Shipbuilding Corporation
J. F. Duthie & Company
Federal Shipbuilding Corporation
Ford & Kendig Company
Great Lakes Engineering Works—
    Detroit Plant
    Ashtabula Plant
Groton Iron Works
M. W. Kellogg Co.
Manitowoc Shipbuilding Company
McDougall-Duluth Co.
Merrill-Stevens Shipbuilding Corporation
Merchant Shipbuilding Corporation
Newport News Shipbuilding & Dry Dock Company
New York Shipbuilding Corporation
Pennsylvania Shipbuilding Company
Stephen Ransom, Inc.
Saginaw Shipbuilding Company
Seattle Construction & Dry Dock Company
Solvay Process Company—Solvay and Detroit Plants
Southwestern Shipbuilding Co.
Standard Shipbuilding Corporation
G. M. Standifer Cons. Corp.
Staten Island Shipbuilding Company
Sun Shipbuilding Company
Toledo Shipbuilding Company
The Panama Canal
Western Pipe & Steel Company
Whitlock Coil Pipe Company

LOVEKIN PIPE EXPANDING & FLANGING MACHINE CO.

421 Chestnut Street Philadelphia, Pa.

MARINE AUXILIARIES



Reilly Type "D" Heater

Evaporators	Lubricating Oil Coolers
Distillers	Fuel Oil Heaters
Steam Condensers	Steam Separators
Grease Extractors	Oil Strainers

The Griscom-Russell Co.

2125 West Street Building

New York

JOURNAL

OF THE

AMERICAN SOCIETY OF NAVAL ENGINEERS

VOL. XXXI. AUGUST, 1919. No. 3.

The Society as a body is not responsible for statements made by individual members.

THE PASSING OF THE DIRECT-CONNECTED TURBINE FOR THE PROPULSION OF SHIPS.

By C. W. Dyson, Rear Admiral, U. S. Navy, Member.

In 1897 Sir Charles Parsons read before the British "Institution of Naval Architects" an article on "The Application of the Compound Steam Turbine to the Purpose of Marine Propulsion." During the ensuing discussion he made the following statement: "Down to fifteen knots screws should be driven direct; below that gearing would have to be used, and then the turbine would be applicable to all classes of ships." He looked with distrust on gearing; the small gearing might be made to work, but the application of large geared wheels was a questionable device.

The mountain peak from which Sir Charles viewed the future was not of sufficient height; his horizon was limited. It did not include either electric or hydraulic gears nor, what is of still more importance, did it include the present day requirements for powers of such magnitude that the fitting of direct

connected slow speed turbines for the development of these powers is prohibited by the enormous sizes of turbines which would be required and the practical difficulties which would be encountered in manufacturing and in fitting them on board ship.

In the few years since the first disclosure of the invention of mechanical reduction gear as installed on board the *Vespasian,* the application of the invention has extended from the modest power requirements of that vessel to the enormous demands for power of the British cruisers *Courageous* and class and the still higher powered *Hood.*

In about the same period of time electric propulsion has grown from the small initial installation in a Chicago fire tug to the 30,000 shaft horsepower of the *New Mexico,* with the contemplated 60,000 shaft horsepower in the latest designed battleships and 180,000 shaft horsepower in the battle cruisers.

Hydraulic reduction gear is a German invention and its development has been seriously interfered with by the Great War. Up to the breaking out of the war the largest installation made was that on the *Koenigen Louise,* a vessel which was sunk off the coast of England early in the struggle, while engaged in laying mines.

CONDITIONS PRODUCING NEED FOR "REDUCTION GEARS."

When the axial flow turbine first made its appearance in the field of marine propulsion, the turbine was connected directly to the line shafting driving the propellers. A necessary condition for turbine efficiency is that the revolving buckets or blades shall have a very high peripheral velocity. On the other hand, the requirements necessary to produce high propeller efficiency are low peripheral velocity of propeller blades, narrowness of blade widths and large propeller diameters; these latter requirements mean that, for high propeller efficiency when absorbing any given amount of power in a propeller, the revolutions of the propeller must be comparatively low.

It will be seen from the above that the requirements for turbine efficiency and for propeller efficiency are directly antagonistic.

The designers of the original turbine installations for marine propulsion deliberately neglected the question of propeller efficiency to such an extent that the whole problem of economy of propulsion was so stated to the public as to eliminate the question of the propeller from the problem and make it appear that the turbine economy was the only factor to consider in solving the problem for "economy of propulsion."

When the direct connected turbine is fitted in any vessel which operates through a wide range of speeds and whose ordinary operating speed and power are much below the designed full speed and full power, the loss in economy becomes much increased, as the steam consumption of the turbine per shaft horsepower delivered increases as the power being developed and the revolutions of the turbine decrease, slowly at first but increasing rapidly as the amount of these decreases in revolutions and power increases, until at very low percentages of power as compared with designed power the steam consumption becomes enormous.

This peculiarity of the turbine becomes of the utmost importance when considering the question of the type of machinery to adopt for propelling purposes in the cases of fighting ships. These vessels in addition to very high power and high speed requirements must also be capable of holding the sea for long periods of time without the necessity for refueling as frequency of refueling is a factor of vital military importance.

When merchant vessels are considered this question of decrease in economy of the turbine need not be considered, as such vessels steam at or near the designed speed at all times when in free route and the engines are therefore developing designed or close to designed power, so that the full power economy is being approximately realized in actual service under usual service conditions.

In the case of war vessels, therefore, it is necessary to provide such an arrangement of machinery as will render good economy not only at the designed speed and power but also at the small fraction of this power which is required for the low cruising speeds usually employed.

For both vessels of war and for merchant ships, however, any steps which can be taken to increase the general economy of propulsion are compulsory when means for doing this are existent and the object of this paper is to outline the steps which have been taken to date to increase the economy of propulsion of war vessels and the step common to both vessels of war and of commerce for the general betterment of this economy.

COMPARATIVE STEAM ECONOMIES OF TURBINES AND RECIPROCATING ENGINES.

In making a comparison of the economies of the above types of engines, as the typical Naval reciprocating engine the engines of the *Delaware* have been taken although they are not the best of their type, those of the *Texas* and *New York* and, of still later date, those of the *Oklahoma* surpassing them in economy under similar steam conditions.

As typical turbines, those of several destroyers, some with Curtis and some with Parsons turbines, have been taken, although by doing so the reciprocating engine is again handicapped as higher economies of the machine are realized in the small high-speed destroyer turbines than can be attained with the heavy battleship engines of the same type.

The reciprocating engines chosen have a ratio of H. P. to L. P. cylinder of about 1 to 9, a ratio of expansion of about 11, a steam pressure at the throttle of 265 pounds per gage and a vacuum in the condenser of 27 inches, the piston speed at designed power being 1,000 feet per minute.

The turbines operate with a steam pressure of 250 pounds per gage at the throttle and a vacuum of 28.5 inches of mercury.

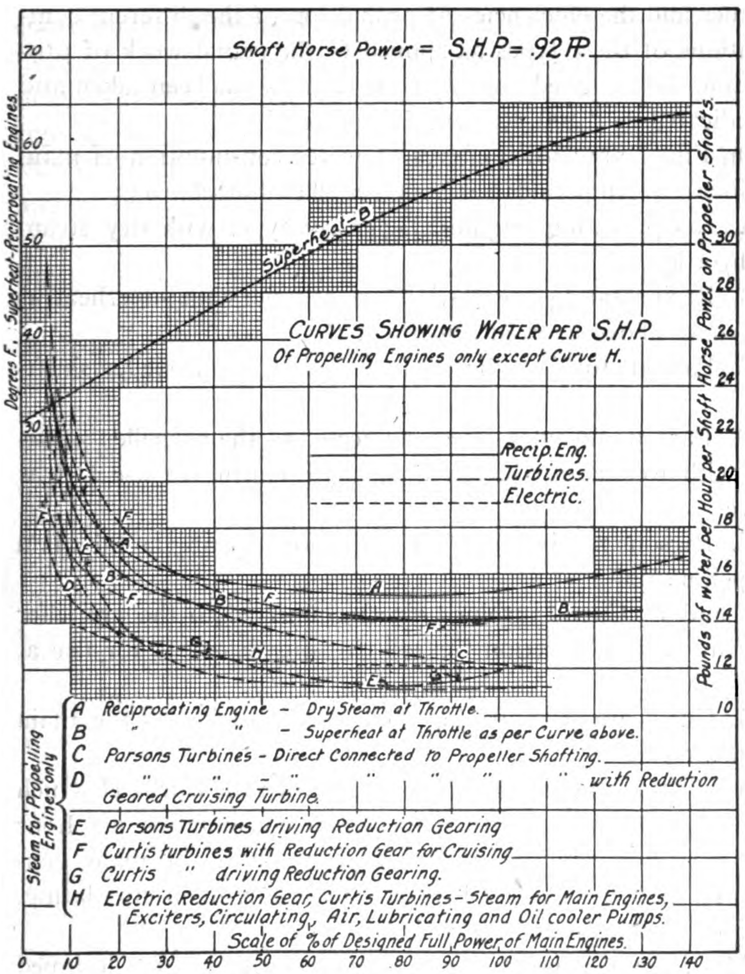

Shaft Horse Power = S.H.P = .92 I.H.P.
CURVES SHOWING WATER PER S.H.P
Of Propelling Engines only except Curve H.
Recip. Eng.
Turbines.
Electric.
Superheat-B
Degrees F., Superheat-Reciprocating Engines.
Pounds of water per Hour per Shaft Horse Power on Propeller Shafts.
Steam for Propelling Engines only
A Reciprocating Engine - Dry Steam at Throttle.
B " " - Superheat at Throttle as per Curve above.
C Parsons Turbines - Direct Connected to Propeller Shafting.
D " " - " " " " " with Reduction Geared Cruising Turbine.
E Parsons Turbines driving Reduction Gearing
F Curtis turbines with Reduction Gear for Cruising
G Curtis " driving Reduction Gearing.
H Electric Reduction Gear, Curtis Turbines - Steam for Main Engines, Exciters, Circulating, Air, Lubricating and Oil cooler Pumps.
Scale of % of Designed Full Power of Main Engines.

FIG. 1.

BASIC POINT OF COMPARISON.

In order to compare the efficiencies of the various types of engines and the efficiencies of propulsion of the different combinations of these types as applied to the actual work of propulsion, the designed power of the engines has been taken and is indicated as "100 per cent full power."

On Fig. 1 are shown curves of water consumption of main engines only, in pounds per hour per shaft horsepower:

A. Reciprocating engines, *Delaware* type, with dry steam at throttle.

B. Reciprocating engines, *Delaware* type, with superheat as per curve above marked Superheat B.

C. Parsons turbines direct connected to the propeller shafting.

D. Parsons turbines direct connected to the propeller shafting with Reduction Geared Cruising Element for use at low powers.

E. Parsons turbines driving Reduction Gearing as the main propelling unit.

F and F. Curtis turbines direct connected to the propeller shafting with Reduction Geared Cruising Element for use at low powers.

G. Curtis turbines driving Reduction Gearing as the main propelling unit.

In addition to the above curves, is shown the Curve H which gives for electric propulsion the water consumption per hour per shaft-horsepower on the propeller shaft, of main generators, exciters, electric boosters, main air, main circulating, forced lubrication and oil cooler pumps.

All of these curves are erected on percentages of "designed full power of the engines" as abscissas while the ordinates are "pounds of water per shaft horsepower on the propeller shafts" and degrees, Fahr., of superheat.

By examination of curves A and B it will be found that where superheat is applied, even in such small amounts as

shown by the superheat curve given, the advantage derived is of such great amount as to be worthy of the most serious consideration. Thus for the case in question, the gain in water consumption at 30 per cent full power, with 41 degrees superheat, is 6.7 per cent; at 50 per cent of full power with 46.3 degrees superheat, the gain has risen to 7 per cent; at 100 per cent of full power with 58.6 degrees, the gain is 8 per cent. While B represents actual *Delaware* conditions, A represents, approximately, those of the *Oklahoma*, superheaters not having been fitted to that vessel on account of certain difficulties existing at that time, with the result that the best reciprocating engines ever designed in the Bureau of Steam Engineering of the Navy Department are caused to appear on the face of the returns as decidedly inferior to the earlier *Delaware* engines, which, again in turn, are decidedly inferior to those of the *Texas* and *New York*, which latter engines in their factors of design resemble very closely the engines of the *Oklahoma*.

The gains given above are greater than the actual gain in economy, as the figures given do not take into consideration the additional heat required to produce the superheat. An average rough figure for the gain due to superheating may be taken as one per cent gain for every ten degrees of superheat.

Turning now to curves C and F and comparing them with A and B, it is seen that C, the direct-connected Parsons turbine, is superior in economy to the reciprocating engine using dry steam from approximately 17 per cent of full power throughout the ranges of power above this percentage, while it is only superior to the reciprocating engine with superheat as shown, from 33 per cent and above. The curve F, the direct-connected Curtis turbine, is superior to the reciprocating engine with dry steam from 33 per cent of full power and upward, while it is superior to that engine with superheat as shown, in the ranges of power above 80 per cent of full power. Attention is called to the character of the curves A, B, C and F. A and B both show a decrease in water consumption more or

less rapid until 80 per cent of full power is being developed. At this point the water consumption gradually increases due to the rapid increase of back pressure in the L. P. cylinders caused by the restricted area of the exhaust ports. C and F show no such changes but the water consumption per S. H. P. is decreasing rapidly throughout the entire range of powers given, indicating an excess of area for steam flow through the turbines, and this decrease would continue so long as the peripheral bucket speed increased and the areas for steam flow could take care of the amount of steam flowing without a banking up of resistance sufficient to require an undue expenditure of pressure to overcome it.

MEANS ADOPTED TO OBTAIN INCREASE IN ECONOMY AT LOW FRACTIONS OF DESIGNED FULL POWER.

Inspection of curves A, B, C and F show that with direct connected turbines there are two remedies which can be applied to reduce the steam expenditures at low fractions of designed power, without introducing an additional propelling element to be used at such powers. These remedies are:

1. The use of superheated steam.
2. By designing the turbines so that the curves of steam consumption C and F will be moved bodily to the left. This would require much higher steam and bucket velocities at the 100 per cent power design point than were used with the turbines from which these curves were derived.

The increase in economy of the engines due to the use of superheat is shown on Fig. 1, curves A and B, where with only 58.6 degrees Fahr. superheat at the 100 per cent design point, the decrease in steam consumption due to the superheat amounts to 7 + per cent, while at 10 per cent of the designed power, the gain, with only 34.5 degrees superheat, amounts to 3¾ per cent.

The gains obtained by resorting to the second remedy are shown by the comparison of curve G with curve F for Curtis

turbines, and of curve E with curve C for Parsons turbines. In the case of the Curtis turbine, in applying the second remedy, gain in economy at the 100 per cent point has been deliberately sacrificed in order to increase the gain at the lower fractions of power so that while at the 100 per cent point the steam consumptions of F and G are rising, it is seen that the maximum gain shown by G occurs at 75 per cent of full power, this gain amounting to about 20 per cent, while at 10 per cent of designed full power it amounts to approximately 35 per cent. G, however, considering the engine economies only, is superior to A and B throughout all practical ranges.

Curves F and G are, however, not the best that the Curtis turbine can deliver as will be seen by examination of curve H, Fig. 1. This curve shows the performance of a Curtis turbine used for driving the generator of an electric drive installation. The bucket speeds averaged about 5 per cent higher than those of curve G, while the superheat ranged from about 15 degrees Fahr., at 10 per cent of designed power to 24 degrees Fahr., at designed power. The water consumptions of curve H, however, in addition to the consumption for the main turbine, include the consumptions for the following auxiliaries, of which only the first two given are a legitimate charge against the engines:

1. Exciters.
2. Electric boosters.
3. Main circulating pumps.
4. Main air pumps.
5. Lubricating pumps.
6. Oil cooler pumps.

Auxiliaries 1 and 2 are extra auxiliaries required for electric propulsion, while the remainder are common to all systems of propulsions although 6 is not usually fitted in cases A and B.

Turning now to the Parsons turbine performances as shown by curves C and E, the latter curve shows a water consumption at full power about 9 per cent lower than C, nearly 7 per cent

better than G, approximately 21 per cent better than B and nearly 27 per cent superior to A. At the points of maximum economy for G, B and A, the superiorities of E in economy are approximately 15 per cent, 20 per cent and 26 per cent respectively. At 10 per cent of designed power the superiorities of E over B and A are approximately 7 and 11 per cent in the order given.

Correcting curve H for the steam required for auxiliaries 3, 4, 5 and 6, would cause it to drop down more nearly to the position of E at the higher percentages, causing it to cross E at a still higher value of percentage of full power than shown, which is about 29 per cent, below which value electric propulsion turbines are far superior to either the marine Curtis or the Parsons turbines in steam economy, while the application of high degrees of superheat, which can be safely used with this turbine but cannot be used with the Parsons turbine, unless impulse first stage is fitted, would put the electric propulsion as well as the marine Curtis turbines far in the lead in economy.

APPEARANCE OF CRUISING ELEMENTS.

In the first steps taken to improve the economy of propulsion at the lower percentages of power and in some cases, throughout the range of powers up to full power, neither remedies 1 nor 2 were resorted to. Instead of these remedies, that of fitting other steam driven machines for use throughout a limited range of the lower percentages of power, these auxiliary units being either disconnected at, or running idle in a vacuum throughout, the higher ranges, was adopted. These variations of auxiliary units, classed as remedy

1. Fitting of Cruising Elements—were adopted in the following order:

a. Adopting of cruising turbines; these usually consisted of a H. P. and an I. P. cruising turbine with their steam areas and bucket speeds more nearly adjusted to the economy re-

quirements for the speeds and powers at which they were to be used than were those of the main turbines at these same powers. In the case of Curtis turbines, these auxiliary turbines were not used but in place of them special cruising nozzles were fitted to the main turbine. The H. P. cruising turbine or cruising nozzles were, in the case of a 21 knot battleship, kept in operation up to about 15 knots; the I. P. cruising turbine or nozzles kept in up to about 19 knots; from 19 knots on the main turbines or main nozzles only were used, the cruising turbines turning idly in a vacuum. The cruising turbines when in operation exhausted to the main H. P. turbine.

b. Adoption of reciprocating engines for cruising in place of cruising turbines; such a plan was first tried in some destroyers of the British Navy, but the mistake was made of designing these engines for such a low power and speed that they were practically useless. Later when adopted in the U. S. Navy, they were designed to carry the destroyer up to fifteen knots and when properly designed were very successful.

c. Combination of reciprocating engines and L. P. turbines for increase in economy throughout the entire range of designed power; this combination was first adopted in some merchant vessels where the power was arranged on three shafts, the outboard shafts each being driven by a triple expansion reciprocating engine with both engines exhausting to one L. P. turbine on the center shaft. The combination was very successful but as designed was not suitable for naval work where the ordinary cruising speeds are far below the maximum speed of the ship. For such a case it is necessary to use the H. P. and I. P. cylinders only of a triple expansion engine on one shaft, these highly efficient cylinders taking the place of the comparatively inefficient H. P. turbine, while the inefficient L. P. cylinder is replaced by the highly efficient L. P. turbine. If the beam of the ship is sufficient to carry propellers of diameter enough for high propulsive efficiency, the arrangement as outlined should give economies of propulsion in excess of those given by either reciprocating or direct connected tur-

bine engine installations except at such fractions of designed speed and power where the L. P. turbine becomes a drag on the system. So long, however, as the L. P. turbine is developing power, the combination will be much more economical in steam consumption than either of the direct connected installations, reciprocating or turbine, operating singly. This combination as outlined was proposed for installation in two of the U. S. dreadnaughts contracted for in 1909, and bids were invited for such installations, but circumstances were such at that time that it was considered better not to make the experiment. Several years later, however, the French installed such an arrangement in several of their new vessels.

d. Adoption of cruising turbines with reduction gear; trials of the *Vespasian* and of the U. S. S. *Neptune* having demonstrated the reliability and efficiency of reduction gears when properly designed, built and installed, and when absorbing comparatively low powers, such gears were immediately turned to as the solution when the cruising radius speed of U. S. destroyers was raised from 15 knots to 20 knots, this latter speed requiring entirely too high a speed of revolution for reliability if reciprocating engines were used. As the direct connected cruising turbines had given way to the much more efficient cruising reciprocating engine, so the latter was forced to make its exit and the unit of cruising turbines transmitting their power to the main shafts through reduction gears made its appearance.

No data being available which gives the water consumptions for the propelling engines only, when fitted with direct connected cruising reciprocating or turbine engines, the following data which includes the water consumption for auxiliary machinery also, and which has been obtained from the trial reports of similar vessels is given in order that a good idea of the relative economies of direct connected turbines in combination with cruising elements as outlined in *a, b* and *d* may be obtained:

	Direct connected turbines with	Fraction of Designed Power	Pounds of Water per S. H. P. per hour for all purposes
a.	Direct connected cruising turbines	.047	35.486
b.	Direct connected cruising recip. engines..........	.04	29.828
d.	Cruising turbines connected through reduction gear...	.04	34.543

Inspection of the above table indicates an enormous gain in economy of steam when the change from *a* to *b* was made but that a decided loss occurred when the change from *b* to *d* occurred. As already stated, however, in order to obtain reliability of machinery at high cruising revolutions the change was necessary even though the loss in economy came with it.

Data of performances of the *d* combination are available and such performances, that is, the pounds of water per shaft horsepower per hour, excluding auxiliaries, are shown on Fig. 1, where the short curve F-F at the left of the figure gives the water consumption for Curtis turbines fitted with *d* for cruising while the curve D is the same for Parsons turbines fitted with the same type of cruising element. The Curtis combination is seen to have the advantage in economy below 9 per cent of full power. At 9 per cent of full power the water consumptions of the two types are each 17.2 pounds while at 5 per cent, the Curtis turbine combination requires 19.8 pounds and the Parsons 22.4 pounds; but both are inferior in economy to H, electric, at the lower fractions of power while the Curtis marine combination as given by the combination of the two curves F-F, F-F is much inferior throughout the entire range of powers. As for the Parsons E curve, D shows more economy throughout the range where used, while the Curtis combination, F-F is only superior below 14 per cent of designed power.

To summarize the steam economies of the different types, the following table showing the standing in the order of economy is given:

Type	Percentage of Designed Power		
	5	30	Full
Reciprocating Engine—Dry Steam	7	6	7
Reciprocating Engine—Superheat	6	4	6
Parsons Turbines—D. C.	8	5	4
Parsons Turbines—D. C. with G. C. E.	4	...	...
Parsons Turbines—Reduction Gear	5	2	1
Curtis Turbines —D. C.	9	7	5
Curtis Turbines —D. C. with G. C. E.	3	...	...
Curtis Turbines —Reduction Gear	2	3	3
Electric Propulsion	1	1	2

The above results show the relative economy standing of the propelling engines considered as machines only and take no account of propulsive efficiencies of propellers that can be realized nor, therefore, of the combined economies of propulsion of engines and propellers. This point of view will now be considered.

ECONOMY OF PROPULSION, ENGINES AND PROPELLERS.

PROPULSIVE EFFICIENCY OF PROPELLERS.

In applying the foregoing types of machinery to the actual work of propulsion, a vessel of the Dreadnaught type has been chosen, while the propeller performances of the *Delaware, Pennsylvania* and *Utah* have been taken as representing the best that may be expected under the three conditions considered, namely, the *Delaware* at 125 revolutions, *Pennsylvania* at 214 revolutions and the *Utah* at 310 revolutions. The standard vessel used in the comparison has an effective (towrope) horsepower and speed curve as shown on Fig. 3. The effective horsepower for 21.3 knots, the designed speed is 15,250.

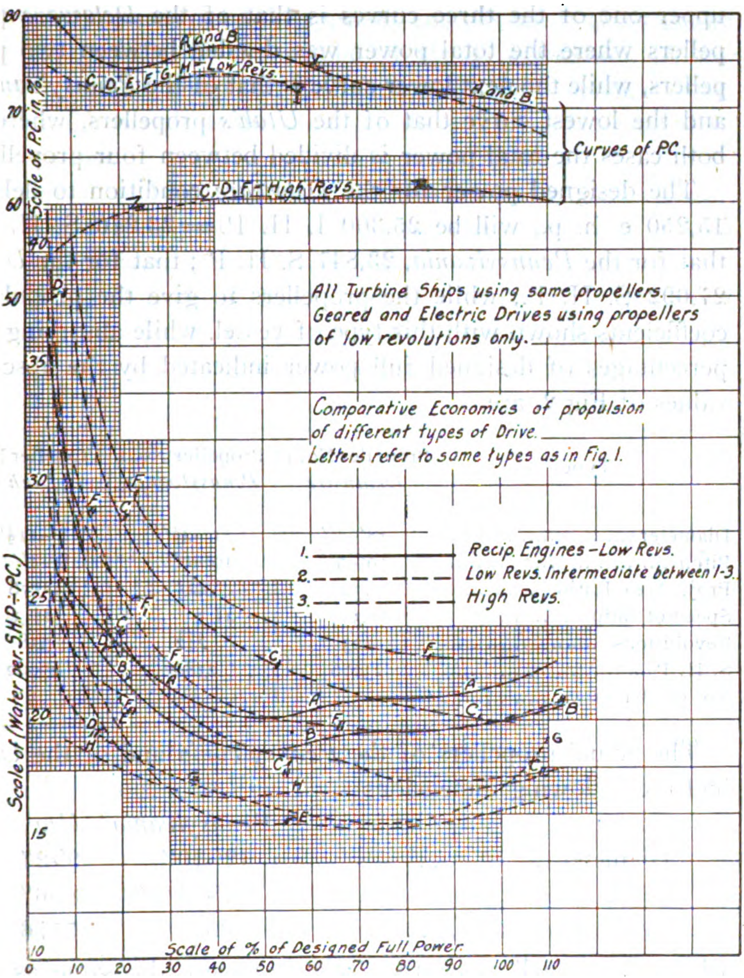

A and B
C, D, E, F, G, H - Low Revs.
A and B.
Curves of P.C.
C, D, F - High Revs.
Scale of P.C. in %
All Turbine Ships using same propellers.
Geared and Electric Drives using propellers
of low revolutions only.
Comparative Economics of propulsion
of different types of Drive.
Letters refer to same types as in Fig. 1.
1. Recip. Engines - Low Revs.
2. Low Revs. Intermediate between 1-3.
3. High Revs.
Scale of (Water per. S.H.P - P.C.)
Scale of % of Designed Full Power.
0 10 20 30 40 50 60 70 80 90 100 110

Fig. 2.

On Fig. 2 are shown three curves of propulsive coefficients laid down on percentage of designed power as abscissas. The upper one of the three curves is that of the *Delaware* propellers where the total power was divided between two propellers, while the next lower curve is that of the *Pennsylvania's* and the lowest curve that of the *Utah's* propellers, where in both cases the total power is divided between four propellers.

The designed power for the *Delaware* condition to deliver 15,250 e. h. p., will be 25,300 I. H. P. = 23,276 S. H. P.; that for the *Pennsylvania,* 23,847 S. H. P.; that for the *Utah,* 27,092 S. H. P., while the propellers to give the propulsive coefficients shown with this type of vessel, while absorbing the percentages of designed full power indicated by the abscissa values of Fig 2, are

Type	Propeller No. 1 *Delaware*	Propeller No. 2 *Pennsylvania*	Propeller No. 3 *Utah*
Diameter	18′-3″	11′-4½″	8′-11¼″
Pitch	19′-9″	10′-0″	8′-4″
Proj. Area Ratio	.32	.425	.5576
Speed of Ship	21.3	21.3	21.3
Revolutions	125	242	338
S. H. P	23276	23847	27092
No. of Blades	3	3	3

The actual propellers of the *Pennsylvania* and of the *Utah* had the following dimensions:

	Pennsylvania	*Utah*
Diameter	12′-10″	9′-2″
Pitch	11′- 3½″	8′-6″
Proj. Area Ratio	.425	.5576

while the propellers of the *Delaware* were the same as for Propeller No. 1.

TRUE MEASURE OF ECONOMY; COMBINED ECONOMY OF ENGINES AND PROPELLERS.

The true measure of economy of propulsion for any vessel with any particular type of propelling machinery as compared

with the performance of a similar vessel equipped with a different type of engine, is not the measure and comparison of the steam economies of the types of engines but is the measure and comparison of the combined steam economies of the engines and the propulsive efficiencies of the propellers when driving the vessels at the same speeds under similar conditions of resistance.

To make such a comparison for the Standard vessel chosen, Fig. 2 has been prepared as a preliminary step towards the final comparison. The curves shown carrying the same letters of the alphabet as Fig. 1, are the values of "Pounds of Water per S. H. P. per hour ÷ Propulsive Coefficient at the same percentage of total designed power," where

F_1 = F ÷ *Utah* P. C.
F_{11} = F ÷ *Pennsylvania* P. C.
C_1 = C ÷ *Utah* P. C.
C_{11} = C ÷ *Pennsylvania* P. C.
E = E ÷ *Pennsylvania* P. C.
G = G ÷ *Pennsylvania* P. C.
H = H ÷ *Pennsylvania* P. C.
A = A ÷ *Delaware* P. C.
B = B ÷ *Delaware* P. C.

and the letters preceding the equality sign refer to Fig. 2, and those following it to Fig. 1. By Fig. 2 is shown the relative costs per hour per S. H. P. at equal percentages of designed full power when using propellers No. 1, No. 2 and No. 3 which represent approximately the existing practices for propeller design for the various types of machinery under discussion. Such comparison does not satisfy when a direct comparison of the relative economies to be expected with any given vessel is sought in making the decision as to which type of propelling machinery will be the best to install, considered from the point of steam economy per knot traveled.

As the second step towards obtaining the desired comparison, Fig. 3 has been prepared. This Figure gives the

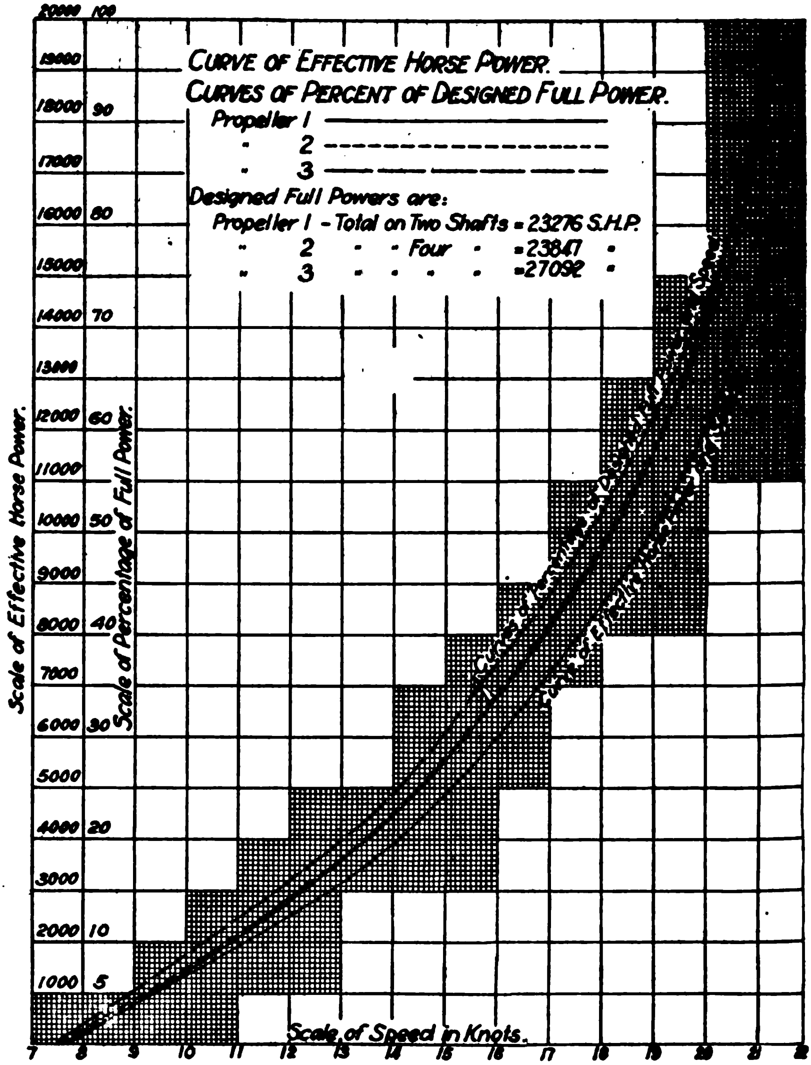

CURVE OF EFFECTIVE HORSE POWER.
CURVES OF PERCENT OF DESIGNED FULL POWER.
Propeller 1
" 2
" 3
Designed Full Powers are:
Propeller 1 - Total on Two Shafts = 23276 S.H.P.
" 2 " " Four " = 23847 "
" 3 " " " " = 27092 "
Scale of Effective Horse Power.
Scale of Percentage of Full Power.
Scale of Speed in Knots.
1000
2000
3000
4000
5000
6000
7000
8000
9000
10000
11000
12000
13000
14000
15000
16000
17000
18000
19000
5
10
20
30
40
50
60
70
80
90
7
8
9
10
11
12
13
14
15
16
17
18
19
20
21
22

Fig. 3.

" effective horsepower and Speed" curve for the Standard vessel and also gives the curves of " Percentage designed full power and speed" curves for propellers No. 1, No. 2 and No. 3.

By combining Figs. 2 and 3, Fig. 4 is obtained. This Figure shows directly the relative economies of engines and propellers for the Standard ship when running at the speeds given by the scale of abscissas. It must be borne in mind fully that the values given by the scale of ordinates are not pounds of steam per shaft horsepower but are such values divided by the propulsive coefficients which are usually obtained with the type of hull possessed by the Standard vessel and the different propellers which are fitted according to current practice.

Tabulating the results given by Fig. 4, for 21, 19, 15 and 10 knots, and giving 100 per cent as the efficiency of that combination which is the best at any given speed, also allowing an average correction of five per cent for electric propulsion in order to give an approximation to the actual economy of that system by allowing this percentage, which is small, for the cost of the main air, main circulating, forced lubrication and oil cooler pumps, we have:

No.	Letter	Type	Propeller No.	Measure of economy per cent. at Speed Knots			
				10	15	19	21
1	F-3	D. C. Curtis	3	43.1	59.6	66.2	70
2	C-3	D. C. Parsons	3	46.5	64.3	72.1	78.2
3	F-3-C	1 with Geared Cr.	3	56.1	67.5	66.2	70
4	D-3-C	2 with Geared Cr.	3	59.4	80	72.1	78.2
5	F-2	D. C. Curtis	2	55.5	71.4	78.2	80
6	C-2	D. C. Parsons	2	56.1	77.1	85	90.8
7	F-2-C	5 with Geared Cr.	2	76.2	77.8	78.2	80
8	D-2-C	6 with Geared Cr.	2	73.2	90	85	90.8
9	E-2	Parsons Geared	2	66.9	93.9	100	100
10	G-2	Curtis Geared	2	91.6	94.5	97.5	96.9
11	H-2	Electric Propulsion	2	100	100	96.9	97.8
12	A-1	Recip. Dry Steam	1	70.2	75	76.7	74.3
13	B-1	Recip. Sup. Steam	1	71 2	80	82.4	80

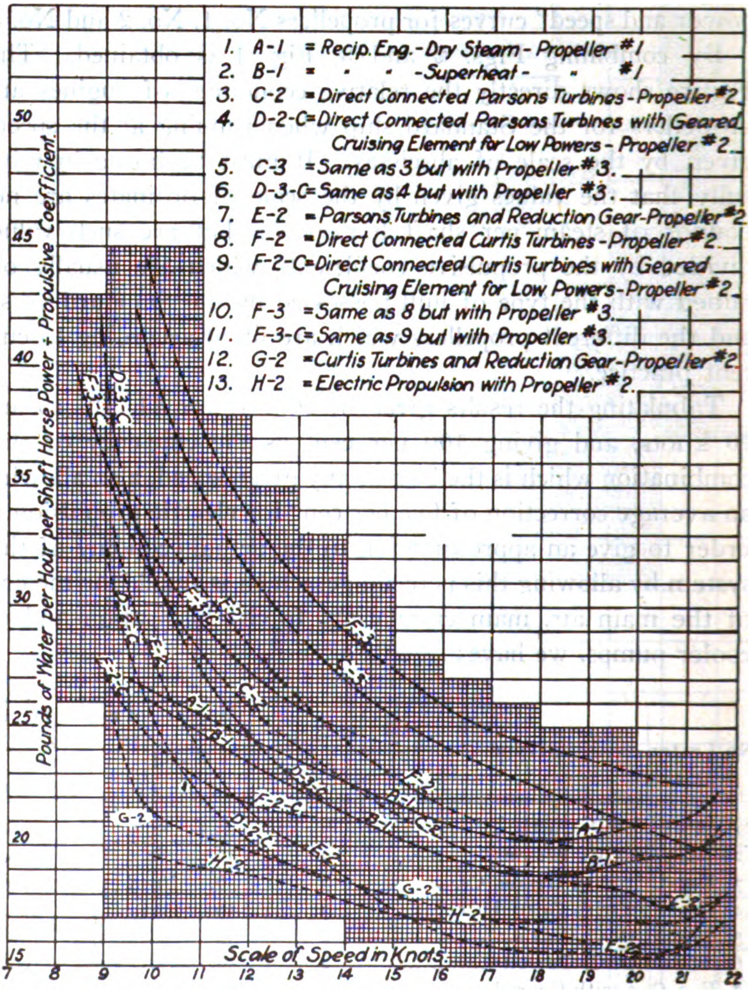

1. A-1 = Recip. Eng. - Dry Steam - Propeller #1
2. B-1 = " " - Superheat - " #1
3. C-2 = Direct Connected Parsons Turbines - Propeller #2
4. D-2-C = Direct Connected Parsons Turbines with Geared Cruising Element for Low Powers - Propeller #2
5. C-3 = Same as 3 but with Propeller #3.
6. D-3-C = Same as 4 but with Propeller #3.
7. E-2 = Parsons Turbines and Reduction Gear-Propeller #2
8. F-2 = Direct Connected Curtis Turbines - Propeller #2
9. F-2-C = Direct Connected Curtis Turbines with Geared Cruising Element for Low Powers - Propeller #2
10. F-3 = Same as 8 but with Propeller #3.
11. F-3-C = Same as 9 but with Propeller #3.
12. G-2 = Curtis Turbines and Reduction Gear - Propeller #2
13. H-2 = Electric Propulsion with Propeller #2.
Pounds of Water per Hour per Shaft Horse Power ÷ Propulsive Coefficient.
Scale of Speed in Knots.
15
20
25
30
35
40
45
50
7
8
9
10
11
12
13
14
15
16
17
18
19
20
21
22

FIG. 4.

On examination of the foregoing Table and of Fig. 4, there is seen to exist a large difference between the economies of the Parsons direct connected turbine installations and those in which the Curtis marine turbine, direct connected, are used, with the difference decidedly in favor of the Parsons installations except where geared cruising elements are used. When such elements are fitted the Curtis installation with the medium speed propeller No. 2 is seen to be slightly superior from 10½ knots down. Both Parsons and Curtis installations with geared cruising units are superior in economy to the Parsons Geared installation at speeds below 14½ to 15 knots, but are both inferior to the Curtis geared at all speeds.

Furthermore, it is seen that the earlier superiority of the reciprocating engine practically disappeared when the propeller conditions for the other types of drives were changed from those represented by Propeller No. 3 to those of Propeller No. 2, and geared cruising elements were adopted.

With the adoption of the Direct Geared drive another advance in economy of propulsion was realized, that with Parsons turbines at the highest speed and at the high cruising speed of 19 knots standing number 1 on the list, being far superior to all the other types except electric propulsion and the Curtis Geared at these speeds, but being inferior at 10 knots to both number 7 and 8 conditions of the Table and being far inferior to condition numbers 10 and 11 of the same Table.

It is seen that condition number 11 is far superior in economy at all speeds up to and slightly above 15 knots, to all other types of machinery, yet at the higher speeds it is apparently very slightly inferior to both Geared drives. This inferiority is, however, so small that it is completely outweighed by the other advantages possessed by the Electric Drive and can be disregarded in making choice between the two types.

As to the Curtis marine turbine which shows up so well in the above comparison, its resort for assistance in raising its economy of propulsion to a still higher plane appears to be high superheat which it can use with a much greater degree of safety than can the Parsons turbine unless the latter be fitted with an impulse high pressure stage, a change which is bound to come in the next few years on this side of the water as it has already come on the other.

NAVY ESTIMATES OF THE PAST.

BY GEORGE W. BAIRD, REAR ADMIRAL (AND FORMERLY CHIEF ENGINEER), U. S. N., RETIRED, MEMBER.

The great demonstration which took place on "preparedness day" in 1916, was a public declaration that "the American people" were urging appropriations for the National defense, and it re-kindled a desire, of the writer, to look up the estimates, and see if any of the neglect was due to "our people" of the Naval Service.

Old men live much in the past, and take satisfaction in defending their acts, even if they are "ancient history."

The search verified the fact that the estimates, for appropriation, as presented by the War Department and the Navy Department, including all the Bureaus, from the time of the Revolution, had clearly in view the strengthening of the Nation's defenses, and though these estimates were often "cut" in Committee and in the Body of Congress, there was evidence that it was done in the interest of economy, which is easily understood when it is remembered that so great a majority of "our people" live a long way from the sea board and a longer way from any belligerent power.

Before we had any outlying possessions to defend there was no apparent reason for increasing the Navy.

Urgent economy made it imperative to dismantle the Navy, discharge the personnel and lay up or sell the ships after the war for independence. This spirit of economy, though "penny-wise" prevailed until "preparedness day" in 1916.

In this search into "ancient history" the writer found some things of interest.

In the estimates for the year 1802, when the ships were appropriated for separately, there was more money asked for and appropriated for "spirits" than for beef and pork to-

gether, and we had pretty good men then. For one Frigate there was estimated 284 barrels of beef, at $13 = $2,692; 244 barrels of pork, at $18 = $4,392. And there was estimated 7,118 gallons of spirits, at $1.125 = $8,007.75.

We had four Frigates, of 44 guns each.
three of 36 guns each.
two of 32 guns each.
two of 32 smaller guns each.
one Schooner.

The larger Frigates had, each, one Captain; four Lieutenants; one Surgeon; one Purser; one Chaplain; one Sailing Master; two Surgeon's Mates; one Boatswain; one Sailmaker; one Carpenter; sixteen Midshipmen; two Master's Mates; two Boatswain's Mates; eleven Gunner's Mates; two Carpenter's Mates; one Captain's Clerk; one Coxswain, a Yeoman, a Cooper, a Steward, an Armorer, a Master-at-arms, a Cook.

The Captain was allowed eight rations, the Lieutenants three rations, etc., but the Petty-officers and crew only one ration. The grog ration was either whiskey, brandy, gin or rum, and was served twice a day, in "tots" of about one-and-a-half fluid ounces. The Ward-room officers' grog was usually placed in a pitcher, on the side-board, and was regarded as common Ward-room property. There were many officers who never touched it, but they never questioned the inherent right nor taste of their ship-mates.

The personnel, of that day, is of interest. All of the artificer class, as they are now called, were *bona fide* mechanics, having served indentured apprenticeships, as was the custom in that day. A "Jack-knife Carpenter" would not be tolerated. The Carpenters were *bona fide* ship-wrights; the Armorer a skilled smith, capable of making and tempering, dressing and grinding his own tools, and able to do any job in smithy required on board. The guns were all smooth-bore, muzzle-loaders, as simple as could be made, and the modern officer may ask why employ eleven gunner's mates?

There was a spirit of chivalry in the old Navy, inspired, no doubt, by that great man, John Paul Jones, who, in his advice to the Continental Marine Committee, specified:

"It is, by no means, enough that an officer of the Navy should be a capable mariner: He must be that, of course, but also a great deal more. He should be, as well, a gentleman of liberal education, refined in manner, punctillious courtesy, and the nicest sense of personal honor. He should not only be able to express himself clearly and with force in his own language, both with tongue and pen, but should be versed in French and Spanish.

"He should be the soul of tact, patience, justice, firmness and charity.

"No meritorious act of a subordinate should escape his attention or be left to pass without its reward, if even the reward be only one word of approval. Conversely he should not be blind to a single fault in any subordinate though, at the same time, he should be quick and unfailing to distinguish error from malice, thoughtlessness from incompetence, and well meant shortcoming from heedless or stupid blunder.

"As he should be universal and impartial in his rewards and approval of merit so should he be judicial and unbending in his punishment or reproof of misconduct."

SOCIETIES, ASSOCIATIONS AND COMMISSIONS PROMULGATING SPECIFICATIONS FOR ENGINEERING MATERIALS.

By K. D. Williams.

There are a number of societies, associations and commissions which have various functions utilized by Naval Engineers. In the course of planning and executing Naval Engineering work it is often necessary to consult and conform to standards, rules and regulations promulgated and enforced by many of the above. For the benefit of such Naval Engineers the following is written with the hope that it may prove of use as an index when special information or specifications are required.

The following list of societies, associations and commissions have been found during recent years to be the nearest associated with the Naval Engineer:

(*a*) Steamboat-Inspection Service, Department of Commerce, Washington, D. C.

(*b*) Bureau of Standards, Department of Commerce, Washington, D. C.

(*c*) American Bureau of Shipping, 66 Beaver Street, New York, John W. Coutillion, Secretary and Treasurer.

(*d*) American Society for Testing Materials, C. L. Warwick, Asst. Sec'y, University of Pennsylvania, Phila., Pa.

(*e*) American Society of Mechanical Engineers, W. E. Bullocks, Asst. Sec'y, 29 West Thirty-ninth St., New York, N. Y.

(*f*) Bureau of Explosives, Underwood Building, 30 Vesey St., New York, N. Y., J. E. Fairbanks, Sec'y and Treas.

(*g*) American Institute of Electrical Engineers, 33 Vesey St., New York, N. Y., F. L. Hutchinson, Secretary.

(*h*) Master Car Builders' Association, 746 Transportation Bldg., Chicago, Ill., V. R. Hawthorne, Acting Secretary.

(*i*) American Railway Engineering Association, Room 1011, 910 Michigan Avenue, Chicago, Ill., E. H. Fritch, Secy.

(*j*) American Institute of Weights and Measures, 20 Vesey St., New York, N. Y., Frederick A. Halsey, Commissioner and Secretary.

STEAMBOAT-INSPECTION SERVICE.

The Steamboat-Inspection Service is charged with the duty of inspecting vessels, the licensing of the officers and crew of vessels and the administration of the laws relative to such vessels and their officers for the protection of life and property.

The Supervising Inspector General and the Supervising Inspectors constitute a Board that meets annually at Washington, and establishes regulations for carrying out the provisions of the Steamboat-Inspection laws.

Publications.

1. *Laws Governing the Steamboat-Inspection Service.*

2. *General Rules and Regulations,* prescribed by the Board of Supervising Inspectors:

(*a*) Ocean and Coastwise.

(*b*) Great Lakes.

(*c*) Lakes other than the Great Lakes, Bays; and Sounds.

(*d*) Rivers.

3. *Pilot Rules:*

(*a*) Inland Waters of the Atlantic and Pacific Coasts and the Coast of the Gulf of Mexico.

(*b*) Western Rivers and the Red River of the North.

(*c*) Great Lakes.

Legal Status.

The act of Congress approved July 7, 1838, which provided for the inspection of the hulls and boilers of steam vessels, was the first legislation upon the important question of "the

better security of lives of passengers on board of vessels propelled in whole or in part by steam." By this act the inspectors were appointed by the district judge of the United States for the district in which the vessel was inspected, and were paid by the steamer the sum of $5.00 each for each inspection. This act provided for lifeboats, signal lights, fire pumps and hose, and also for iron rods or chains in place of rope for steering gear. The boilers were inspected every six months and the hulls every twelve months, and it was also required that steamers employ "a competent number of experienced and skillful engineers." Certificates of inspection were required "to be posted up and kept in some conspicuous part of the boat for the information of the public."

This act was modified by the act approved March 3, 1843, which provided for additional steering apparatus, but exempted certain vessels from the requirements of the previous act. The act of Congress approved March 3, 1849, provided for signal lights for all vessels.

The act of Congress approved August 30, 1852, was really the establishment of the present Steamboat-Inspection Service, and was known as the "steamboat act." This act amended the act of 1838, and provided for the appointment and payment of nine supervising inspectors, and for local inspectors at various ports. The act also provided for lifeboats, life-preservers, and other life-saving equipment, and for the licensing of engineers and pilots for passenger steamers, and for the stamping of boiler plate, and is really the foundation upon which the whole superstructure of the Steamboat-Inspection Service has been raised.

This act was modified and amended in minor particulars by the various acts of March 3, 1853, April 29, 1864, June 8, 1864, March 3, 1865, and July 25, 1866.

The act of Congress approved February 28, 1871, gave what is really the present law upon the organization and administration of the Steamboat-Inspection Service, and although amended and changed from time to time it is, in fact,

the basis of the present practice. This act provided for the appointment of a Supervising Inspector General and created the Board of Supervising Inspectors, designating also the time and place of meeting of said Board and defining the duties thereof. It provided for the inspection, testing, and stamping of material entering into the construction of marine boilers, a provision which has proven of the highest value and importance in the prevention of explosion and loss of life.

The bill upon which this legislation was based was prepared with much care, and its enactment was, without doubt, the wisest legislation ever devised upon these lines. It has served well the purposes for which is was designed and has fully justified the highest expectations entertained by those who supported it.

Legal Authority to Establish Standards or Conduct Inspections.

Under the provisions of Section 4405, R. S., the Supervising Inspectors and the Supervising Inspector General are required to assemble as a Board once a year in the City of Washington, District of Columbia, on the third Wednesday in January, and at such other times as the Secretary of Commerce shall prescribe, for joint consultation, etc., and that Board is required to establish all necessary regulations required to carry out in the most effective manner the provisions of Title 52.

Under the authority of Sections 4417 and 4418, R. S., the hulls and equipment and boilers of vessels with appurtenances are required to be inspected by the Boards of Local Inspectors.

Section 4438, R. S., gives Boards of Local Inspectors the authority to license and classify officers of vessels.

Section 4450, R. S., gives the Boards of Local Inspectors authority to investigate all acts of incompetency or misconduct committed by any licensed officer while acting under the authority of his license.

BUREAU OF STANDARDS.

The National Bureau of Standards was established by Congress to have the custody of standards, comparison of standards, preparation of standards, and the solution of problems which arise in connection with standards.

The standards with which the Bureau is authorized to deal may be conveniently classed as follows: Standards of measurements, standards of values of constants, standards of quality, standards of mechanical performance, and standards of practice.

Publications.

The Bureau of Standards publishes four series of publications:

(*a*) *Bulletins* which comprise the separate scientific papers covering researches of the Bureau within its field.

(*b*) *Technologic papers.*

(*c*) *Circulars* covering information of a technical or administrative character.

(*d*) *Miscellaneous publications.*

Legal Status.

The Organic Act of March 3, 1901, established the Bureau of Standards by stating that "the office of standard weights and measures shall hereafter be known as the National Bureau of Standards," and the same act expanded its functions to accord with the modern needs for standardization.

Successive acts such as the Legislative Appropriation Act, the Sundry Civil Act, and others have provided the necessary authority and funds for specific investigations.

Legal authority to establish standards and specifications as noted above was established by an act of Congress and supplemented by Section 3 of the Organic Act authorized the Bureau

to exercise its functions for the federal, state, or municipal, governments, societies, institutions, firms, corporations, or individuals.

AMERICAN BUREAU OF SHIPPING.

Quoting from the Constitution and By-Laws:

"The objects of the Bureau are as shown by the charter, and for the purpose of carrying out these objects the Bureau is to provide for shipowners, shipbuilders, underwriters, shippers, and all interested in maritime commerce, a faithful and accurate classification and registry of mercantile shipping, and to aid and develop the Merchant Marine of the United States of America."

The Bureau maintains a corps of inspectors whose duty is to inspect material both in the process of manufacture and fabrication and to inspect completed vessels and machinery; classification certificates are issued on the findings of these inspections.

Publications.

The Bureau publishes annually a "*Record of American and Foreign Shipping.*" This "Record" for the year 1918 has more than 1,100 pages, giving the following information:

List of Offices, where situated, etc.

List of Officers, Managers, Committees, Counsel, etc.

List of Surveyors and Ports. In this connection it may be said that there are Surveyors throughout the United States and in Foreign Ports all over the world.

List of Subscribers to the Record.

List of Vessels, including signal letters, name, rigging, nation, hailing port, registered dimensions, tonnage, when and by whom built, owners, engine and boilers, and memoranda as to classification, surveys, damage surveys, etc.

List of Owners and vessels owned by same.

List of Shipbuilders, Dry Docks and Marine Railways and Constructors of Marine Machinery and boilers throughout the world.

The Bureau also publishes semi-monthly or as often as needed, supplements to the "Record," giving particulars of new vessels and changes to old ones.

The Committees referred to above under "List of Officers" are: Executive, Finance and Audit, Naval Architecture, Engineering and Classification.

Legal Status.

The Bureau was incorporated under the laws of the State of New York, April 22, 1862, under the Title of "American Shipmasters' Association," and on September 26, 1898, the title was changed by legal authority to "American Bureau of Shipping."

The Bureau under its charter, Constitution and By-Laws examines the plans of vessels submitted for classification, approving same or making suggestions towards complying with the Rules governing the construction; and finally gives a certificate of class and of machinery, and when required assigns the least proper freeboard that the vessel may not be overloaded.

AMERICAN SOCIETY FOR TESTING MATERIALS.

The International Association for Testing Materials was formed at a conference of workers in experimental engineering held in Munich in 1882. Larger and more important meetings have subsequently been held and the membership now extends to thirty-one countries.

It was decided at the Stockholm Congress (1897) to encourage the consolidation of the membership in various countries into separate national organizations. In pursuance of this action the American members met in Philadelphia on

June 16, 1898, and organized under the name of the "American Section of the International Association for Testing Materials."

In 1902 the Association obtained a charter and was incorporated under the laws of the State of Pennsylvania under the name of the "American Society for Testing Materials." The American Society for Testing Materials is represented on the International Council by one member. The membership of the Society in 1918 was 2,261.

Purpose.

One of the most important functions of the Society as stated in its charter, is the standardization of specifications and the methods of testing. In pursuance of this activity the Society has adopted 128 standards covering ferrous and non-ferrous metals, cement, lime, gypsum and clay products, and certain miscellaneous materials such as preservative coatings, lubricants, road materials, rubber materials, rubber products, textile materials, etc. The Society has also published 49 tentative standards which is the term applied to a proposed standard which is printed in the Proceedings for one or more years with a view of eliciting criticism before final action is taken towards their adoption as standards.

Publications.

"*A. S. T. M. Standards*" are published triennially; the next publication will appear in 1921. From 1910 to 1916 the standards were published annually; from 1916 to June, 1918, when publication was put on the present basis, they were published biennially.

In addition to the publications of its standards the Society issued annually *Proceedings* of about 1,200 pages in two parts; Part I contains the committee reports and tentative standards, and Part II the technical papers presented at the annual meeting.

Each of the standards and tentative standards of the Society are available in separate pamphlet form. Any of the above publications may be purchased upon application to the Secretary.

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

The American Society of Mechanical Engineers was founded in 1880 " to promote the arts and sciences connected with Engineering and Mechanical Construction." The Society endeavors to accomplish this object through its meetings, general and section, held periodically through the United States; its publications—The Journal and Transactions—which are distributed free to its members, to libraries in exchange and to others by subscription; its large numbers of committees, working to develop the Society's usefulness in the mechanical engineering profession; and its library, located in New York City, but available to engineers throughout the country through the medium of the Library Service Bureau, a special library mail service.

Publications.

The Journal of the Society is published monthly. Features of the Journal are addresses, papers and discussions presented at general meetings of the Society, addresses, papers or discussions presented at Local Sections, and reviews of technical books by experts selected by the Publication Committee, etc.

The Transactions of the Society is published about July of each year and contains the more important addresses, papers and reports that have been presented at local section meetings during the preceding year.

The Year Book of the Society is published in February of each year and contains a list of its members, the Charter, Constitution, By-Laws and Rules of the Society, etc.

Legal Status.

The Society was organized as a corporation under the laws of the State of New York, April 7, 1880, its office being located in the City of New York.

C 56 of its constitution reads as follows: "The Society shall not approve any standard or formula, or approve any engineering or commercial enterprise. It shall not allow its imprint or name to be used in any commercial work or business."

It maintains a "Committee on Standardization" to formulate and revise standards of principal methods of procedure in Engineering practice. There are upward of fifty standards covering various engineering materials already created and these have been widely adopted and have become the basis of extensive manufactures.

BUREAU OF EXPLOSIVES.

In April, 1905, Mr. James McCrea, later President of the Pennsylvania Railroad, advocated before the American Railway Association the appointment of a committee to prepare regulations to govern the safe transportation of explosives.

This committee was appointed and with the aid of civilian experts in explosives, and consulting representatives of the Army and Bureau of Ordnance, Navy Department, prepared regulations, and the same were approved by the Association in October, 1905. No marked change resulted in the transporation of explosives until the Association in October, 1906, decided that some central agent of all roads was necessary for uniform enforcement of the regulations and a constitution and by-laws for the *Bureau for the Safe Transportation of Explosives and Other Dangerous Articles* were adopted.

In the following year the office of the Chief Inspector was opened in New York City. Seventy-eight companies at that time were members. The present membership comprises 244 railway companies.

At the present time the personnel of the Bureau consists of the Chief Inspector and office force, 23 Local Inspectors, 2 Special Agents, 3 Assistant Local Inspectors, a Chemist, an Assistant Chemist, and a Laboratory Assistant. The duties of the Inspectors are to investigate conditions at manufacturing plants, etc., and make reports to the designated operating officials in their districts and to the Chief Inspector.

The Federal law of May 30, 1908, now codified by the Act of March 4, 1909, as Sections 232 to 236, requires the Interstate Commerce Commission to formulate and the carriers to enforce reasonable regulations to promote safety.

The Bureau of Explosives submitted to the Interstate Commerce Commission a set of regulations for their consideration and with but a few changes were adopted and were prescribed to take effect October 1, 1908.

The Bureau has prepared specifications for containers for the transportation of gases both liquefied and non-liquefied, and files the history and records of their manufacture and test as well as data on containers for the transportation of other dangerous articles.

The cost of maintaining the Bureau is shared by its members, an assessment being levied each six months by the Executive Committee.

AMERICAN INSTITUTE OF ELECTRICAL ENGINEERS.

Quoting an extract from its constitution the purpose of the Institute is stated as follows:

"Its objects shall be the advancement of the theory and practice of Electrical Engineering and of the allied Arts and Sciences and the maintenance of a high professional standing among its members. Among the means to this end shall be the holding of meetings for the reading and discussion of professional papers and the publication of such papers, discussions and communications as may seem expedient."

Publications.

The *Proceedings* are issued at the beginning of each calendar month and have two principal divisions: The first part contains matter of transient interest, such as announcements, election of members, personal notes, reports of the meetings of Sections and Branches, and general information pertaining to the Institute. Technical papers and other matters of an engineering character are contained in the second part.

The *Transactions* are published in one or more bound volumes each year, and consist of technical papers and discussions, reports, and other matter, previously published in the monthly Proceedings, deemed worthy of permanent record.

Legal Status.

The Institute was incorporated on March 16, 1896, under the Membership Corporations Law, Section 5, of the laws of the State of New York.

The Institute does not conduct inspections of any kind. It has established electrical standards, relating to the rating and testing of electrical equipment, but not including specifications of design and dimensions. These standards have been revised from year to year as a result of changes and developments in the electrical art. The Institute is also affiliated with the International Electrotechnical Commission and its standards conform to those adopted by the Commission. These standards, known as the Standardization Rules of the A. I. E. E., are universally accepted as official by engineers throughout the country. The Rules were not formulated under any legal authority, but as one of the proper functions of the national organization representing the electrical profession.

MASTER CAR BUILDERS' ASSOCIATION.

The Master Car Builders' Association was originally composed of master car builders and foremen of railway car shops, and was organized at Altoona, Pennsylvania, September, 1867.

In June, 1881, an amendment to the constitution was proposed, and in October, 1882, this amendment creating a new class of members called "representative members" was adopted. This amendment permitted each railroad to be represented in direct proportion to the number of cars it owned. The number of cars represented in the Association is approximately 2,800,000. Its membership extends to England, France, Russia, India, Australia, Japan, China, Argentina, Chile, Brazil, Cuba and the Philippines.

Purpose.

Quoting from information furnished to the Railroad Administration by the Association's late secretary, Mr. Joseph W. Taylor, the purpose of the Association is as follows:

"The objects of the Association are the advancement of knowledge concerning the construction, maintenance and service of railroad cars and the parts thereof, by investigations through committees and discussions in conventions; to provide an organization through which the members and the companies they represent may agree upon such joint action as may be required to bring about uniformity and interchangeability in the parts of railroad cars; to improve their construction and to adjust the mutual interest growing out of their interchange and repair, but the action of the Association shall have a recommendatory character, and shall not be binding upon any of its members or the companies represented in it."

This Association was responsible for the adoption of automatic couplers for cars in 1887 and the automatic air brake in 1888. Through their efforts also standards for 62 different parts of cars have been adopted.

The standardization of parts of cars and method of loading same is of the greatest value when it is considered how often cars are shifted from one road to another and many repairs are necessarily performed while a great distance from their home shops.

AMERICAN RAILWAY ENGINEERING ASSOCIATION.

Purpose.

The object of the American Railway Engineering Association is the advancement of knowledge pertaining to the scientific and economic location, construction, operation and maintenance of railways.

Publications.

The American Railway Engineering Association publishes:

"*The Monthly Bulletin,*" issued ten times per year (May and June omitted);

"*The Annual Proceedings,*" issued annually;

"*The Manual for Railway Engineering and Maintenance of Way Work,*" issued at intervals of from three to five years. Supplements thereto are issued annually.

Legal Status.

The American Railway Engineering Association is incorporated under the laws of the State of Illinois "not for profit."

Legal authority to establish standards or conduct inspections: The recommendations of the Association are not binding upon the members of companies with which it is connected, but the standards are recommendatory.

The Association does not make "inspections" but has made and is continually making experimental tests of various kinds, the results of which are published for the information and benefit of the members of the organization and railroad companies generally. Such tests cover:

The effect of moving loads on railroad bridges,
Tests on columns (bridge),
Tests of rail joints,
Tests of water columns,
Strength of bridge timbers, etc.

AMERICAN INSTITUTE OF WEIGHTS AND MEASURES.

Quoting from the constitution of the above Institute adopted June 28, 1916, and amended February 17 and 19, 1917, its object is:

"(*a*) The maintenance and improvement of our present (English) system of weights and measures, for the good of our commerce and industry and the well-being of our country.

"(*b*) The education of the people with respect to the importance of our weights and measures, through the dissemination of correct information with respect to them and to the danger inherent in changes of our basic standards of measurement.

"(*c*) The improvement of old and the development of additional standards as they may be needed by reason of new conditions in commerce, industry, science, and engineering.

"(*d*) The promotion of wise legislation for the conservation of our basic English units of weight and measure, and opposition to hasty and ill-considered legislation involving changes from our fundamental English standards."

The Institute publishes a quarterly report of activities issued to members alone. Two Reports on The Metric System in Export Trade and on the Weights and Measures of Latin America have been issued, as well as five bulletins on the following subjects:

Bulletin No. 1—"The Six Metric Myths."

Bulletin No. 2—"Endorsements That Count."

Bulletin No. 3—"The American Metric Association and the Metric System in Munitions of War."

Bulletin No. 4—"The Bushel Standard."

Bulletin No. 5—"Extracts from Reports of British Parliamentary Committees Relating to the Metric System."

The Legal Status of the Institute is that of an unincorporated Society.

RELATIONS OF SOCIETIES TO NAVAL ENGINEERING.

(*a*) *Steamboat-Inspection Service Rules.* Material for use in the Naval Service is seldom purchased in strict accordance with these rules. They have occasionally been used in the past in specifying material for Naval Auxiliary Ships.

(*b*) *Bureau of Standards.* The Navy Department has consulted the Bureau of Standards on new developments where special experimental work has been necessary. During the late war, assistance has been rendered by the Bureau of Standards in investigations and experiments of engineering materials, such as brasses, bronzes and bearing metals, with the view of conserving tin; also in Aircraft materials and gasses.

(*c*) *American Bureau of Shipping.* For vessels of types similar to those mentioned under "American Society of Mechanical Engineers," and for the same reasons, the machinery is often purchased in accordance with the Rules of the American Bureau of Shipping, and other materials are occasionally specified to the above rules.

(*d*) *American Society for Testing Materials.* The specifications of this Society are employed more as references. Navy material such as structural steel for ships and rivets have been purchased in strict accordance with the specifications of this society.

(*e*) *American Society of Mechanical Engineers.* The boilers for such Naval vessels as colliers, oil tankers, repair ships, sea going and Harbor tugs, etc., are ordered often to be in accordance with the A. S. M. E. Boiler Code. This practice is sometimes followed to expedite deliveries and obtain material at a lower cost than if purchased in full accordance with the Navy Department Specifications. As a rule material purchased in accordance with the A. S. M. E. Boiler Code is inspected at the place of manufacture by representatives of the Navy Department.

(*f*) *Bureau of Explosives.* All cylinders or containers for the transportation of gases such as Hydrogen, Oxygen, Helium, Acetylene, Carbonic Acid Gas, Anhydrous Ammonia, etc., are purchased in accordance with the rules or specifications of this Bureau in order that they may comply with the Interstate Commerce rules and be accepted for transportation by the several transportation companies. These rules or specifications are incorporated in the Navy Department leaflet specifications for the containers for the several gases. Containers for the shipment of ammunition and other dangerous articles are also purchased in accordance with the rules and specifications of the Bureau of Explosives.

(*g*) *American Institute of Electrical Engineers.* Motors and controllers which bidders guarantee to give satisfactory operation after installation and certify that they are in full conformity with the A. I. E. E. standardization rules are often accepted for Navy Yard use in lieu of motors and controllers in full accordance with the Navy Department Specifications.

Attention is invited to a clause permitting the above in "General Specifications and Instructions Governing Submission of Proposals for Furnishing Machine Tools for the United States Navy," dated March 15, 1916.

The rules in question are not generally regarded as specifications but are considered an authority in regard to methods of conducting tests, calculating efficiency and other characteristics of motors, generators and transformers; and definitions of various terms used in connection with specifications and tests.

(*h*) *Master Car Builders' Association.* It is necessary for the rolling stock in the various Navy Yards and stations to be in accordance with the rules and specifications of the M. C. B. to insure facilities for the handling of cars belonging to the various railroads and in some cases transporting Navy owned cars over private owned roads.

(*i, j*) *The publications of the American Railway Engineering Association and the American Institute of Weights and Measures* are employed as references by Naval Engineers in charge of creating and modifying Navy Department Specifications.

PROPELLER DATA FROM U. S. S. *NEW MEXICO'S* TRIALS.

By Commander S. M. Robinson, U. S. Navy, Member.

In the February, 1915, issue of this Journal there was published certain data on the subject of propellers in connection with the design of electric machinery for ship propulsion. It has been possible to check some of this data by the recent trials of the *New Mexico* and this article will compare the data of that issue of the Journal with the data obtained from the actual trials of the *New Mexico*. In addition, there will be given certain original information that was obtained in regard to propellers and propulsion generally.

The *New Mexico* held two sets of trials. The first trial was held after the ship had been out of dock about seven weeks. The second trial was held just after the ship had been docked. These two trials have been analyzed and the analysis is given in Table 1. From this it will be seen that there was a difference between the two trials of about 4,250 shaft horsepower at 21 knots, due to the foul condition of the ship's bottom; this increase in resistance, expressed as a percentage, amounted to about 9.6 per cent on the average. This experience of the *New Mexico* was corroborated by that of the *Pennsylvania;* this ship had been out of dry dock for a few weeks and it was decided to try the ship without cleaning her bottom, but a preliminary run demonstrated that the increase of horsepower was so great that it would be impossible to fulfill the guarantee of fuel consumption without docking the ship. It would seem that fouling occurs very rapidly in the first few weeks that a ship is out of dry dock and after that the rate of increase of resistance due to fouling falls off very rapidly.

U.S.S. NEW MEXICO.

HULL DATA		PROPELLER DATA	
Length	= 600′	$\frac{PA}{DA}$	= .3567
Beam	= 97′	P	= 15.167
Draft	= 30′	D	= 13.412
Displacement	= 32000 Tons	EHP	= 21500
$\frac{B}{LWL}$	= .1617	SHP	= 30520
NBC	= .641	V	= 24.85
SBC	= .63	S	= .099
K	= 1		

1st TRIAL

v	$\frac{v}{V}$	$\frac{ehR}{E.H.P.}$	10^{z_1}	$S.H.P_d$	$(10^z)^s = \frac{10^{z_1} \times S.H.P.}{S.H.P_d}$	$10^z = \sqrt{(10^z)^s}$ (with cavitation) or $\frac{SHP}{SHP_d}$ (without)	$\frac{ehp}{EHP}$	ehp (Total)	% Cavitation $= \frac{10^z}{10^{z_1}} - 1$	% increase of ehp due to foul bottom	v^y	w	$1-w$	R_d (Calculated)	R_d (actual)
7	.2815	.04	28.564	1200	726.5	26.95	.044	946	6.0	5.0	459	.1133	.8867	52.7	56.5
10	.4023	.102	10.776	3600	91.4	9.56	.1146	2465	12.75	5.6	1000	.1555	.8445	79.1	81.0
12	.483	.173	6.217	5850	32.43	5.695	.1882	4048	9.1	7.25	1717	.147	.853	94.0	96.1
13.5	.543	.24	4.42	7900	17.07	4.131	.2562	5495	7.0	6.5	2444	.1398	.8602	104.8	107.5
15	.6035	.34	3.075	10500	8.94	2.99	.3483	7510	2.88	10.75	3318	.1367	.8633	116.2	118.3
17	.684	.57	1.796	15300		1.995	.515	11075	0	8.9	4773	.1385	.8615	131.8	135.5
19	.7645	.875	1.149	23300		1.31	.7717	16590	0	11.4	6548	.1537	.8463	150.0	152.0
20.5	.825	.949	1.056	30500	1.057	1.028	.976	20980	2.7	12.10	8081	.163	.837	163.7	164.4
21	.845	.971	1.033	33350	.9455	.9724	1.0266	22080	6.25	7.5	8630	.167	.833	168.5	169.0

2nd TRIAL

v	$\frac{v}{V}$	$\frac{ehR}{E.H.P.}$	10^{z_1}	$S.H.P_d$	$(10^z)^s$	10^z	$\frac{ehp}{EHP}$	ehp (Total)	% Cavitation	% increase of ehp due to foul bottom	v^y	w	$1-w$	R_d (Calculated)	R_d (actual)
7	.2815	.04	28.564	1150	756.0	27.5	.0414	885	3.5		459	.1085	.8915	52.5	56.3
10	.4023	.102	10.776	3200	102.25	10.136	.1085	2333	6.35		1000	.1382	.8618	77.5	78.4
12	.483	.173	6.217	5050	37.55	6.129	.1755	3775	1.4		1717	.127	.873	91.9	93.0
13.5	.543	.24	4.42	6900		4.42	.24	5160	0		2444	.1222	.8728	102.8	104.1
15	.6035	.34	3.075	9175		3.326	.3154	6780	0		3318	.1194	.8806	113.8	115.6
17	.684	.57	1.796	13980		2.182	.4728	10170	0		4773	.1265	.8735	130.0	132.9
19	.7645	.87	1.149	20800		1.467	.6923	14890	0		6548	.1372	.8628	147.2	149.2
20.5	.825	.949	1.056	26400		1.156	.87	18710	0		8081	.141	.859	159.5	161.7
21	.845	.971	1.033	29100		1.048	.9558	20550	0		8630	.1456	.8544	164.2	166.7

Table I.

The ship was also standardized, using only two screws to drive the ship; four points were obtained altogether; three points on the first trials, and one point on the second trials; in each case the points were determined for the inboard and for the outboard screws separately. These trials were run with

DRIVING WITH OUTBOARD SCREWS

v	SHP_d	$\frac{10^{z_1}}{10^z} = 1 + \%$ cavitation	$10^z = \frac{10^{z_1}}{10^z} \times \frac{SHP}{SHP_d}$	$\frac{ehp}{EHP}$	ehp	ehp to drive without dragging screws	Increase of ehp due to dragging screws	ehp to drive without dragging screws and with clean bottom	% increase of ehp due to dragging screws	$10^{z_1} = 10^z \times (1+\%$ cavitation)	$\frac{ehp}{EHP}$	R_d of the driving screws	R_d of the dragging screws	R_d to drive with 4 screws	% R.P.M. of dragging screws
						1st. TRIAL									
7	1660	1.008	9.27	.118	1268	946	322	885	36.4	9.35	.117	76.4	—	56.3	—
10	4325	1.075	3.795	.278	2990	2465	525	2333	22.5	4.08	.259	102.5	47.5	78.4	60.6
13.5	8900	1.02	1.750	.584	6280	5495	785	5160	15.2	1.785	.574	132.4	65.2	104.1	62.6
						2nd TRIAL									
15	11517	1.00	1.325	.7633	8200	6780	1420	6780	20.95	—	—	144.2	73.1	115.6	63.2
						DRIVING WITH INBOARD SCREWS. 1st. TRIAL									
7	1860	1.129	9.27	.118	1268	946	322	885	36.4	10.47	.105	754	—	56.3	—
10	4560	1.133	3.795	278	2990	2465	525	2333	22.5	4.3	.2465	101.4	48.5	78.4	61.9
13.5	9625	1.103	1.75	.584	6280	5495	785	5160	15.2	1.932	.531	131.8	67.2	104.1	64.6
						2nd TRIAL									
15	11654	1.011	1.325	.7633	8200	6780	1420	6780	20.95	1.34	.755	143.5	76.8	115.6	66.4

Table 2.

v	ehp to drive with clean bottom	ehp due to dragging screws	Total ehp	$\frac{ehp}{EHP}$	10^z	$(10^z)^2$	10^{z_1}	$SHP_d = \frac{10^{z_1} \times SHP}{(10^z)^2}$	Rd
				DRIVING WITH OUTBOARD SCREWS.					
7	885	322	1207	.1123	9.763	95.3	9.35	1560	74.6
10	2333	525	2858	.266	3.973	15.8	4.08	3940	97.2
13.5	5160	785	5945	.5525	1.855	3.45	1.785	8220	127.0
				DRIVING WITH INBOARD SCREWS.					
7	885	322	1207	.1123	9.763	95.3	10.47	1680	73.6
10	2333	525	2858	.266	3.973	15.8	4.3	4150	96.5
13.5	5160	785	5945	.5525	1.855	3.45	1.932	8550	126.4

Table 3.

v	SHP_d	R_d	b (actual)	$S \times v^3 = \frac{r \times SHP \times v}{SHP_d}$	SBC	V	S	$\frac{v}{V}$	$\frac{ehp}{EHP}$	10^z	$(10^z)^2 = \frac{10^{z_1} \times SHP}{SHP_d}$	10^z	$\frac{ehp}{EHP}$	ehp (backing) due to slip and propeller.	ehp (backing) due to slip	ehp (backing) due to propeller.	% increase of ehp astern due to propeller	Difference between total ehp astern and ehp ahead in % of ehp ahead
10.2	6720	105.	.352	1704	.4	23.58	.148	.4325	.185	8.734	89.67	6.299	.171	3675	2715	960	37.15	45.5

Table 4.

the motor disconnecting switches open on the idle motors so that the dragging screws were perfectly free to turn. The results of this standardization have been analyzed in Table 2 and plotted as curves in Figure 1. As part of the data was

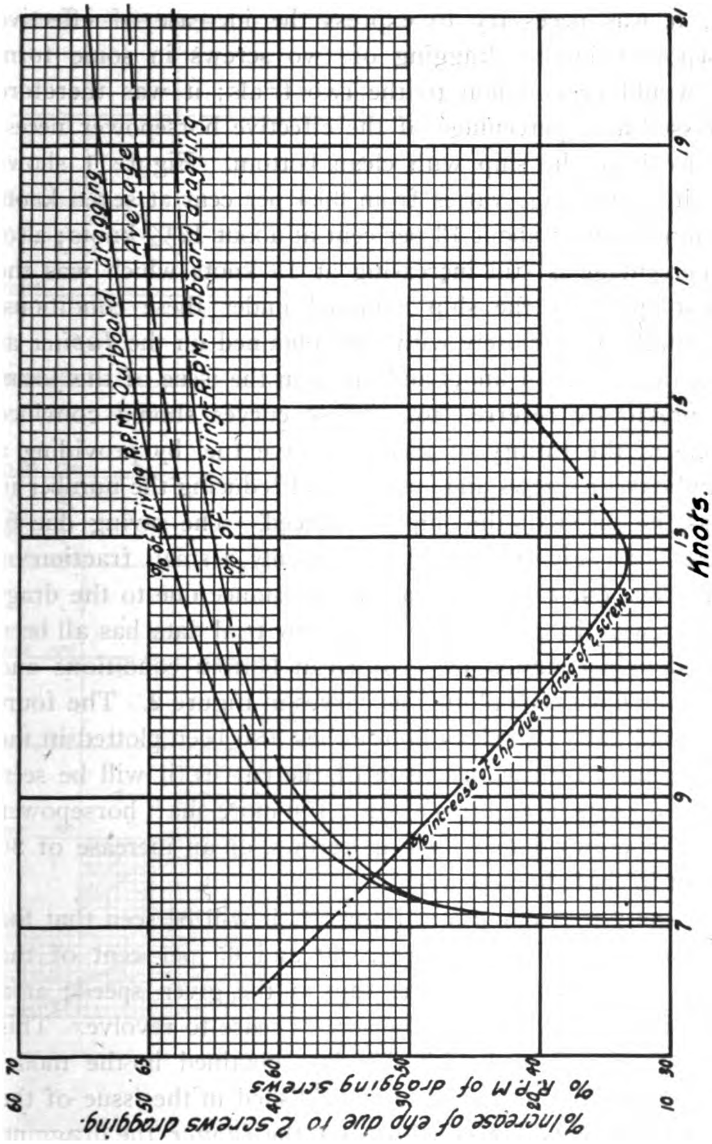

% of Driving R.P.M. Outboard dragging
Average
% of Driving R.P.M. Inboard dragging
% increase of ehp due to drag of 2 screws
Knots.
7
9
11
13
15
17
19
21
% R.P.M of dragging screws
% increase of ehp due to 2 screws dragging.
70
65
60
50
40
30
60
50
40
30
20
10

Fig. 1.

obtained on the trials with bottom clean and part with bottom foul, it was necessary to express the increase of effective horsepower due to dragging of two screws in some form that would be common to the two trials; it was therefore expressed as a percentage of the effective horsepower necessary to drive the ship with clean bottom. Figure 1 shows that this percentage varies from 36.4 per cent at seven knots to a minimum of about 13 per cent at about 12½ knots; also, the percentage is still increasing at 15 knots which was the highest speed of the ship obtained under these conditions. This checks very closely with data obtained on the *Jupiter* at a speed of about 14 knots and given in the issue of the JOURNAL previously referred to. These curves should convince anyone of the futility of trying to save fuel by providing a multiplicity of screws and engines and reducing the number in use as the speed of the ship is reduced. The saving due to increase of engine efficiency will be only a small fraction of what is lost due to the increase of resistance due to the dragging screws. In Table 3 the two-screw trial data has all been reduced to a common basis of clean bottom conditions and the results have been plotted as curves in Figure 2. The four-screw trial data with clean bottom has also been plotted in the same figure. By a comparison of the curves, it will be seen that at 15 knots, it requires about 2,400 more shaft horsepower with two screws than with four screws, or an increase of 26 per cent in shaft horsepower.

Returning to Table 2 and Figure 1, it will be seen that the idle propellers drag at between 60 and 65 per cent of the R.P.M. required to drive the ship at the given speed; at a speed of about seven knots the screws cease to revolve. This data agrees very well with the data obtained in the model tank and also on the *Jupiter* and published in the issue of the JOURNAL previously referred to. On the *Jupiter*, the dragging screws ran at about 68 per cent of the normal R.P.M. required to drive the ship. As this percentage seems to be about the

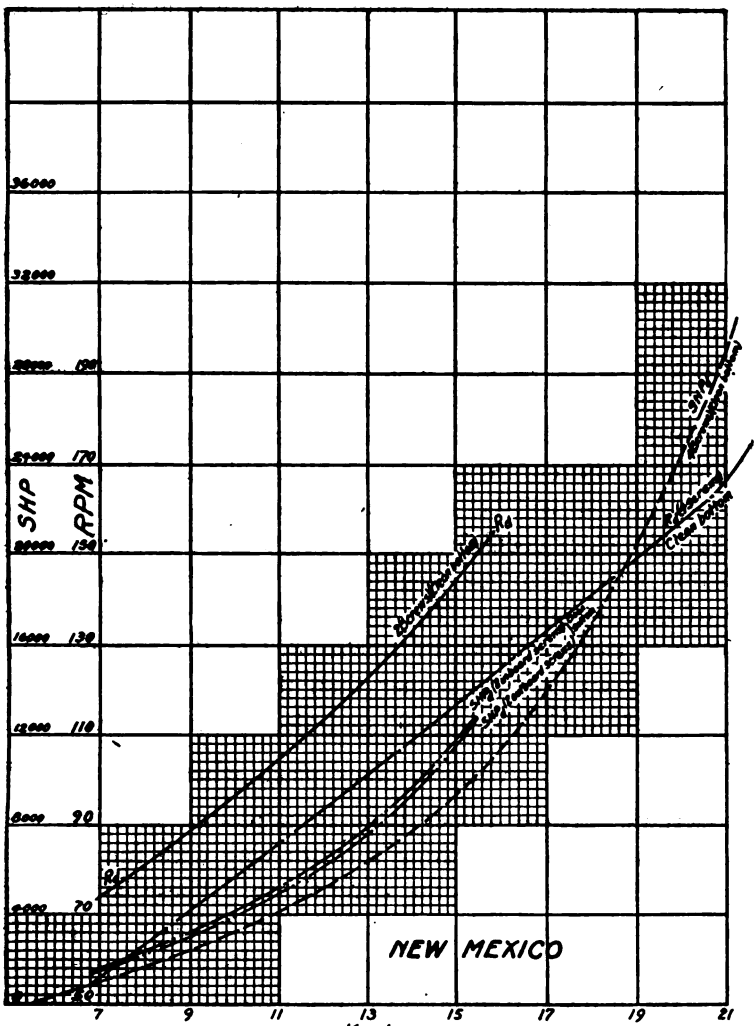

36000
32000
190
170
SHP
RPM
150
16000
130
12000
110
8000
90
70
50
Rd
Clean bottom
NEW MEXICO
7
9
11
13
15
17
19
21
Knots.

Fig. 2.

same for all types of ships, it offers a very easy method of plotting retardation curves simply by observing the revolutions of the engine from instant to instant and obviates the necessity of obtaining the actual speeds of the ship by bearings. It will be noted that when running with two screws, the dragging screws run at approximately one-half of the R.P.M. made by the driving screws.

A comparison of the performance of the inboard and outboard screws, as given in Table 2, shows that more power is required for the same speed by the inboard screws than by the outboard screws, and paradoxically, the R.P.M. of the inboard screws is less than that of the outboard. The apparent paradox in regard to the R.P.M. is easy to understand if we refer to Figure 3 which has been taken from sheet No. 17 of Dyson's "Screw Propellers for Hydraulic and Aerial Propulsion," for it will be seen that there is a small difference in the "slip block coefficient" for inboard and outboard screws and this will, of course, slightly affect the *basic speed and slip* of the two screws. The difference in horsepower is not so easy to explain; it is, of course, due to the fact that the inboard screws cavitate sooner than the outboard. Table 2 gives the critical values of 10^{z1} and also the critical values of $\frac{\text{e.h.p.}_1}{\text{E.H.P.}}$ at which cavitation takes place when driving with two screws. These values have been plotted in Figure 4. The values for the same thing when driving with four screws have also been plotted in the same figure. These last values were taken from sheet No. 22 of Dyson's "Screw Propellers for Hydraulic and Aerial Propulsion." By a comparison of the curves in the two instances it will be seen that on a four-screw ship it is possible, when driving with two screws only, to load these screws very much more heavily before cavitation takes place than can be done when driving with four screws, although the slip of the screw is more than twice as great with the two-screw condition as with the four-screw condition.

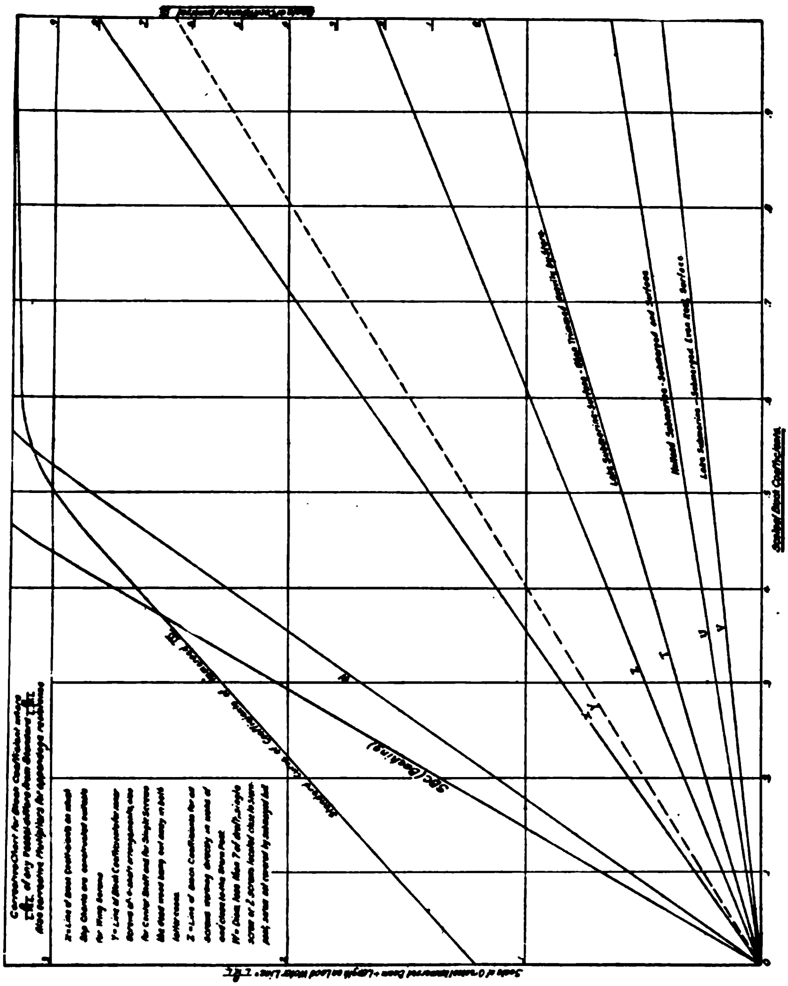


Fig. 3

The ship was also standardized at one speed, when going astern. This data has been analyzed in Table 4. It will be seen that the "slip block coefficient" in this case is quite different from what it is for the ship when going ahead. This value of "slip block coefficient" was found by determining

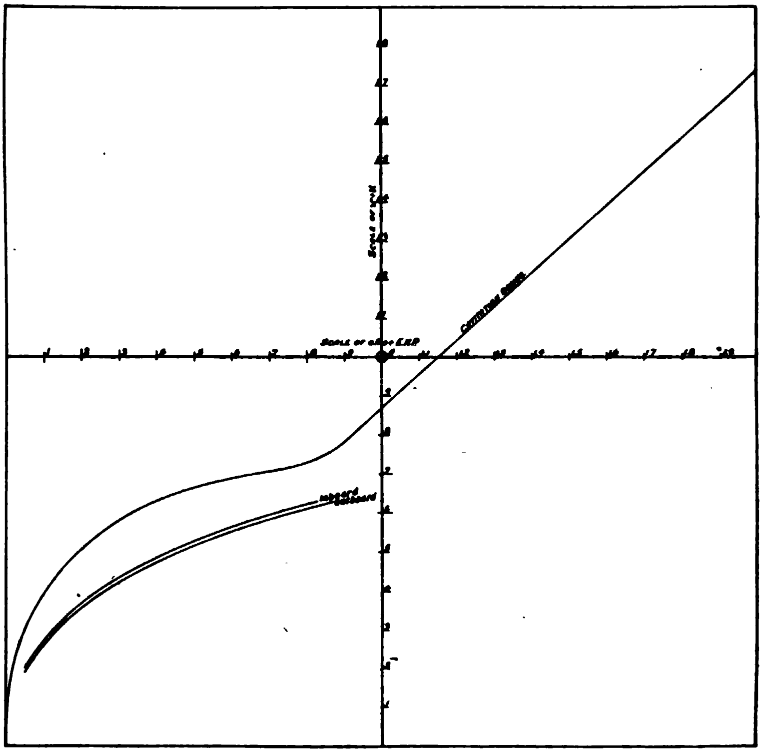


Fig. 4

the basic *speed* (V) from the actual horsepower, revolutions, and slip. The line of "slip block coefficient" as determined by this point is shown in Figure 3. The other "slip block coefficient" lines shown in the same figure have been taken from sheet No. 17 of Admiral Dyson's book. From a comparison of the two, it will be seen that the "slip block" for a ship going astern is radically different from what it is for a ship

going ahead. Table 4 shows very clearly the enormous increase in effective horsepower, due to the reaction of the water thrown against the stern of the ship. In this case it amounts to over 37 per cent of the effective horsepower necessary to tow the ship astern and the total effective horsepower necessary to drive the ship astern is over 45 per cent greater than that necessary to drive her ahead.

In the article in the JOURNAL previously referred to, there was given a curve showing the variations in propeller torque that occur while a propeller is passing from full revolutions ahead to full revolutions astern. This curve was obtained

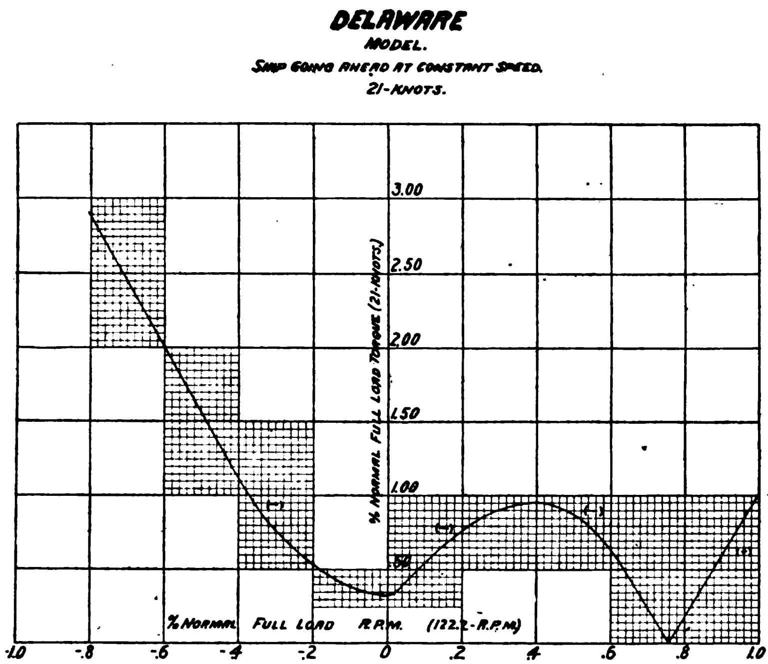


FIG. 5.

from model tank experiments and is reproduced in this article as Figure 5. The most vital part of this curve, so far as it affects induction motor design, is the maximum torque point

of the propeller during the period in which it is being brought to rest. In Figure 5 the value of the torque at this maximum point is about 95 per cent of the torque required to drive the ship at full speed. Data obtained from the *Jupiter* indicated a considerably lower value for this point—about 75 per cent. It was, therefore, very desirable to get a check on the point by the *New Mexico;* the value of the point on the *New Mexico* was found to be about 106 per cent at 18 knots; at the same speed, the model tank curves agree fairly closely, showing about 112 per cent. The value of the torque at this point was obtained by setting the motor switches in the reversing position and gradually building up the field with the main generator until it was just sufficient to reverse the propeller. Continuous graphic records were taken of the speed of the generator, field of the generator, voltage of the generator, and speed of the motor, ampères of the motor and kilowatt input of the motor. These values have been plotted in Figures 6 and 7. From these it was possible to determine the torque at each interval of time from the formula $T = \dfrac{KW \times 33000}{.746 \times 2 \times \frac{\text{Gen. R.P.M.}}{18}}$ The values of torque as determined from this formula are also plotted in Figure 6. It was, however, in addition to this, necessary to know the speed of the ship through the water at the instant that the propeller passed through the maximum torque point; this was obtained from retardation curves made by taking power off the ship and taking continuous readings of the revolutions of the propeller and also bearings of an object by the bridge compass. These curves were obtained with the ship steaming on two opposite courses and the average values of the two runs were used in plotting. This data is shown in Figure 8. To determine the point at which the propeller actually passed through its maximum torque it was assumed that this was when the propeller R.P.M. was 35 per cent of the R.P.M. corresponding to the

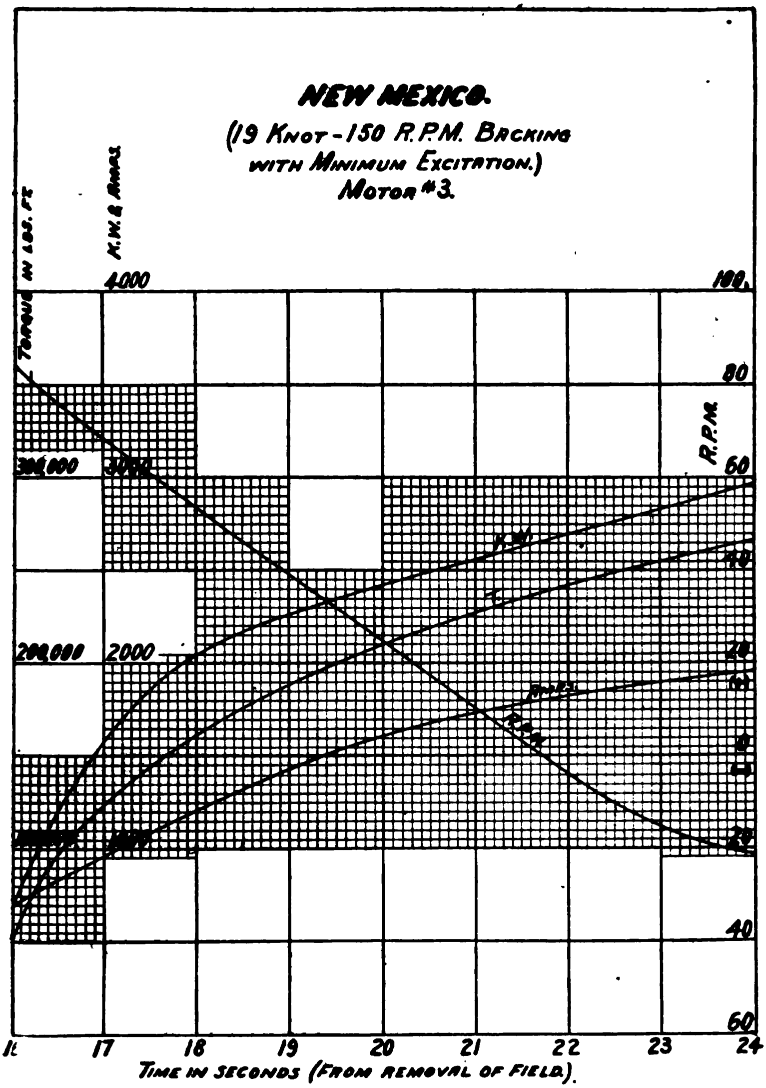

NEW MEXICO.
(19 KNOT - 150 R.P.M. BACKING
WITH MINIMUM EXCITATION.)
MOTOR #3.
TORQUE IN LBS. FT.
K.W. & AMPS.
4000
3000
2000
1000
300,000
200,000
100
80
60
40
20
0
20
40
60
R.P.M.
K.W.
AMPS.
R.P.M.
17
18
19
20
21
22
23
24
TIME IN SECONDS (FROM REMOVAL OF FIELD).

Fig. 6.

speed of the ship, as both *Jupiter* and model tank curves agreed on this. By comparing Figures 6 and 8, a point can

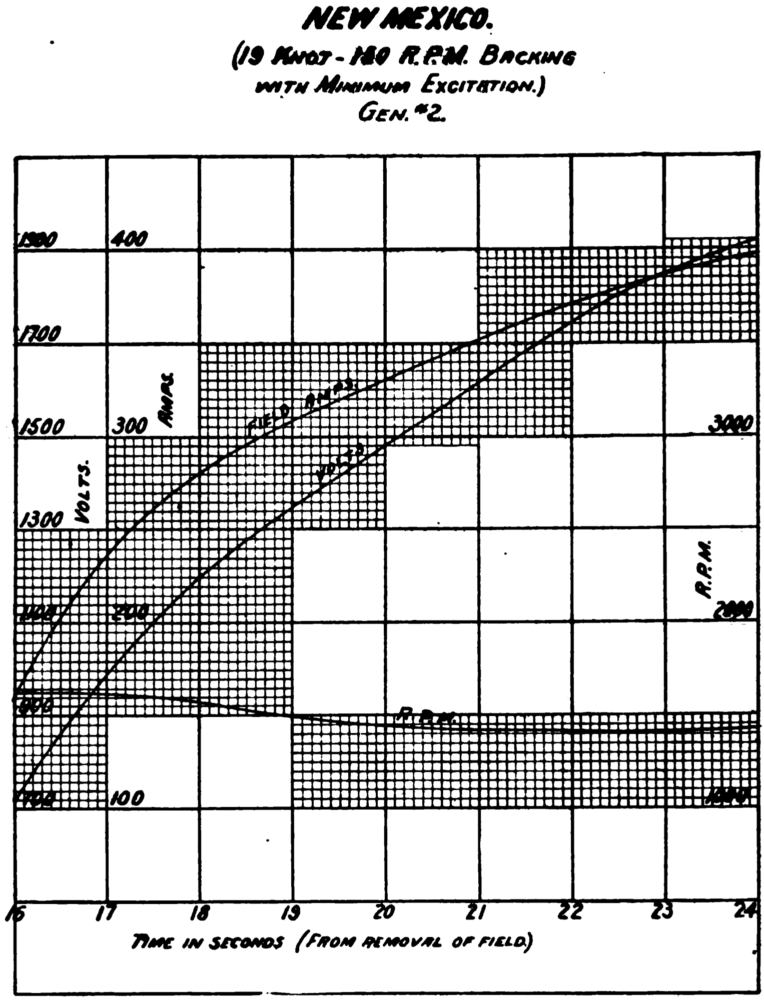


Fig. 7.

be found where the motor R.P.M. are just 35 per cent of the R.P.M. required to drive the ship; the torque at this point,

as given by the curve in Figure 6, will be the maximum torque required to bring the propeller to rest.

The curves in Figure 8 show that it requires about 15 seconds for the propeller to drop down to the actual dragging revolutions corresponding to the speed of the ship. This is also an important point in connection with induction motor design, as it affects the available torque of the motor for reversing. These curves are very similar to curves of the *Jupiter* given in the issue of the JOURNAL previously referred to; in the same issue, there was outlined a method for calculating the deceleration of a ship and a curve was worked out for the *New Mexico,* using this method; this curve is reproduced in Figure 8 as the dotted curve, and it will be seen that it is in very close agreement with the actual curve.

There is one other point in connection with electric propulsion that was discussed in the previous issue of the JOURNAL and that is the "turning of ships." Experiments were made on the *Jupiter* and the *Delaware* by turning these ships and keeping the revolutions per minute of the engines constant. This is the condition that obtains with an engine controlled by a governor. The experiments made on the *Jupiter* and *Delaware* were repeated on the *New Mexico.* The trials carried out on the first two ships were at low speeds, being 12 and 14 knots on the *Jupiter* and 12 knots on the *Delaware.* Turns were made on the *New Mexico* at 14.5 knots, using one generator, and at 19.5 knots, using both generators. On the *Delaware,* the total increase of power due to turning was 39 per cent when using 25 degree of rudder, on the *Jupiter* it was 29 per cent. On the *New Mexico,* at 14.5 knots, the increase was 50 per cent, and at 19.5 knots the increase was 30 per cent, but this does not represent the final limit of increase at 19.5 knots, as the limit of power was reached on the turbine before maximum power was reached. These percentages are affected by the speed of the ship, by the size of the rudder, and by the rudder angle used, so that it is difficult

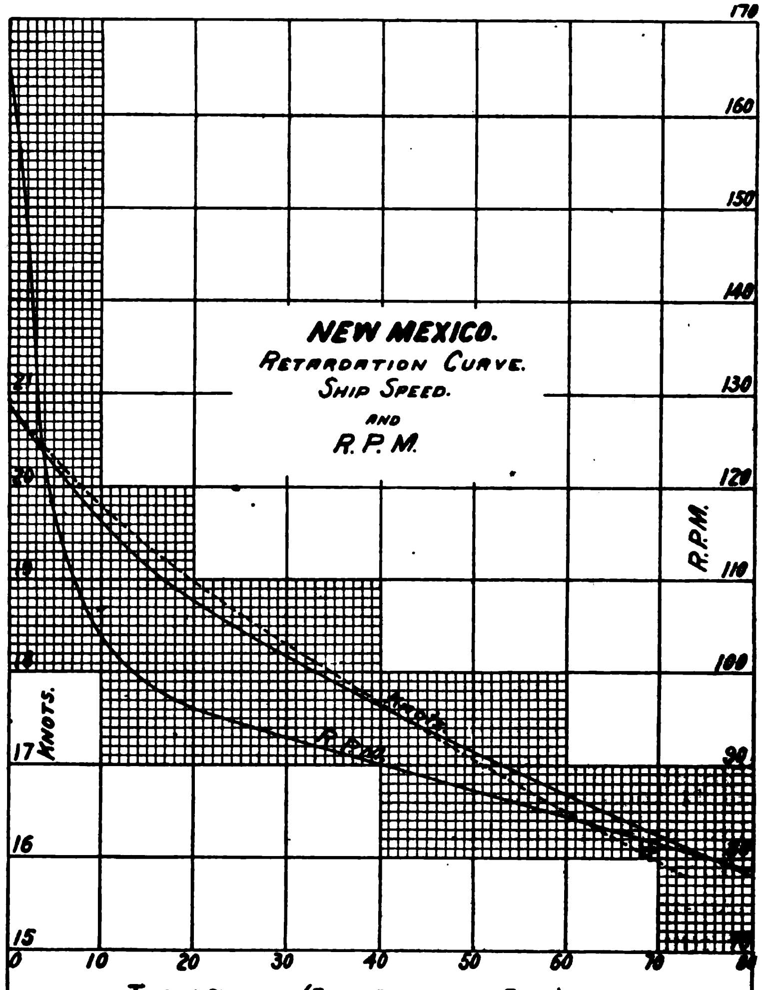

NEW MEXICO.
RETARDATION CURVE.
SHIP SPEED.
AND
R.P.M.
KNOTS.
R.P.M.
R.P.M.
KNOTS.
21
20
19
18
17
16
15
170
160
150
140
130
120
110
100
90
0
10
20
30
40
50
60
70
80
TIME IN SECONDS (FROM REMOVAL OF FIELD)

FIG. 8.

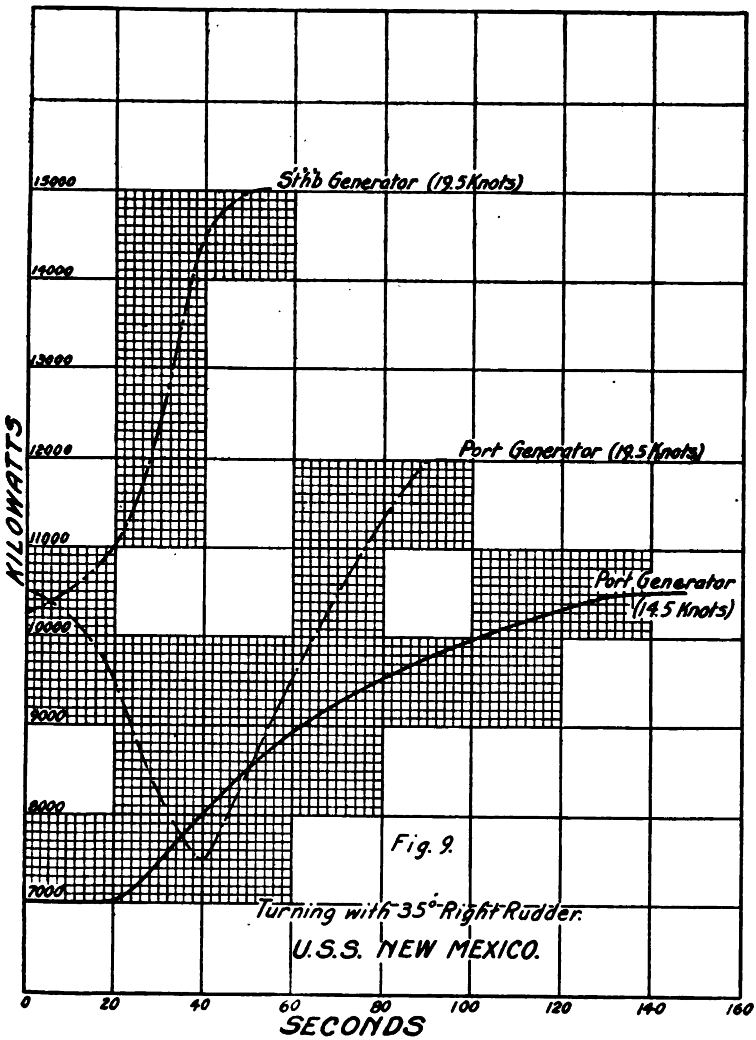


Fig. 9.

Turning with 35° Right Rudder.

U.S.S. NEW MEXICO.

to predict what this percentage will be in any given case. The curves obtained from the *New Mexico's* trials are given in Figure 9. It will be noted that on the 19.5 knot trial the increase of power on the starboard generator was 50 per cent, but this does not represent the limit of the increase, as the maximum power of the turbine was reached before the power rise had stopped; with all valves open on the turbine it was not possible to develop any more power. This condition represents what actually occurs on the *New Mexico* during a turn; the governor is provided with a steam-limiting device which can be set to limit the power of the turbine to any desired amount and if the ship makes a turn, the power rises to this point and remains constant. This device is essential to the proper operation of any electric-driven ship, as otherwise it would be necessary to carry excessive excitation on the generator, thus reducing the efficiency of transmission and increasing the temperatures in the electric machinery.

NOTES ON OPERATION OF SUBMARINE DIESEL ENGINES.

By Lieut. Comdr. Frederick C. Sherman, U. S. Navy, Member.

The following remarks are random notes made while operating under war conditions submarine Diesel engines of 440 H.P., 4-cycle, 6-cylinder type, manufactured by the New London Ship and Engine Co., and installed on the O-class of submarines. As this type of engine has become almost the standard production of that Company and has been installed on a number of our submarines with only variations as to size and number of cylinders, it is believed that this information obtained from early experience with these engines under strenuous operating conditions will be of value to others taking up the operation of the same type engine. In addition, it may suggest ideas which will be of value in operating Diesel engines of other types. This article is somewhat of a continuation of an article by the writer on the same subject published in the Journal of November, 1917.

BEARING TROUBLE.

The O-class of submarines equipped with Nelseco Engines of the 4-cycle type when first placed in commission had considerable trouble with burned out and wiped crank-shaft and crank-pin bearings. No direct cause of this could be discovered, as one or more bearings would go out simultaneously while lubrication was apparently sufficient, everything running normally, and bearing previously adjusted to proper clearance. To understand the lubrication system on these engines a brief description is as follows: The engines

are oiled by a forced lubrication system which comprises a motor-driven rotary oil pump taking oil from a sump or settling tank located in the bilge below the crank case and discharging through a cooler into a pipe with branches to each main bearing, from which the oil passes to the interior of the crank-shaft through the crank web to the crank-pin bearing; then up through the hollow connecting rod to the wrist pin bearing, whence it seeps out and drains by gravity to the crank case from which it further drains to the sump tank. In addition, a mechanical oiler is connected to each cylinder for cylinder wall lubrication, injecting the oil at four different points in each cylinder.

The circumstances of the bearing trouble were as follows: The first indication would be a knock from a crank-pin bearing with subsequent heating of the bearing. Upon opening the crank case, this bearing would be found to be burned out, its babbitt metal melted and run into the crank case. Upon test the bearing would appear to be getting lubricating oil from a good flow through the oil hole in the crank-pin. A visual examination without breaking down of the adjacent main bearings would show no sign of trouble. But upon putting in a new crank-pin bearing, after running an hour or so the bearing would go again. Sometimes the main bearings would be burned out also, but usually the procedure was as above. However, it was found after frequent recurrences of this trouble that even though the main bearings were apparently all right and showed no signs of having been overheated, if they were disassembled and examined they would be found to be slightly wiped and perhaps one of them with the oil hole through the babbitt plugged, so that the crank-pin was only getting oil from one side. This would show a fairly good flow of oil from the crank-pin when stopped but would not give sufficient oil for good lubrication when running. What made this bearing trouble so inexplicable was the fact that bearings would burn out when they had all been carefully ad-

justed to the proper clearance. It would have been perfectly understandable if the bearings had had too much or too little clearance, for too much would cause the oil to wash out and prevent bearings further on from getting sufficient oil, and too little clearance would cause the bearing to run hot and the babbitt metal to flow. But these bearings which were burning out were ones which had been adjusted with extreme care, and they were burning out too regularly to be from careless or faulty adjustment.

The problem then became one of discovering why the main bearings were wiping and shutting off the flow of oil to the crank-pin. In searching for the answer to this question the following theory was evolved:

Frequently when making examination as a result of this trouble, one or more of the oil pipes to the main bearings, or the radial hole in the crank web would be found plugged up with a sticky mass of about the consistency and appearance of coal tar. This occurred in spite of the fact that the lubrication system had been previously thoroughly cleaned out and the fine mesh strainers in the system showed no signs of any foreign matter being caught there. Further investigation showed the inside of the pistons near the top and around the wrist pin coated with carbon of somewhat the appearance of the gummy mass which had plugged the oil holes. This carbon was in an excessive amount and was apparently in all pistons in about equal amount. One piston which was removed as a result of a piston seizure showed heavy carbon deposits inside and also a smaller amount of carbon around the wiper and piston rings on the inside of the piston. Assuming that the carbon forming on the inside of the pistons and that in the lubricating oil pipes was the same, it became necessary to trace how it was formed and how it could get into the lubricating system and through the strainers.

The theory was then evolved that the lubricating oil which seeped out the ends of the wrist pin bearing and drained to

the crank case was being splashed onto the hot piston and cylinder walls and there being evaporated or imperfectly burned, leaving behind a heavy residue. This residue, being hot and still fluid, would be partly scraped off into the crank case and the other part of it would be worked up into the piston rings. This latter part would gradually work on through the rings and then on into the cylinders where it would eventually be burned completely or left as a deposit on the exhaust passages. If sufficient of this residue should be caught in the piston rings at one time, it might be a contributory cause of a piston seizure. As a matter of fact, a number of these did occur coincidently with the bearing trouble. The part of the residue which was scraped off into the crank case, having considerable temperature and fluidity, would flow with the oil through the sump tank and strainers and pump until it came to the cooler. Here the heavy tarry particles would be cooled sufficiently to begin to congeal. But they would flow along with the oil until they struck a restriction or smaller passage such as the pipes to each main bearing or the holes in the crank webs, where the particles would lodge and gradually collect until the oil passage was shut off altogether and that bearing would lose its oil, resulting in the bearing troubles mentioned in the beginning of this paper.

The question then was how to prevent this action, as it was not practicable or desirable to install splash plates to prevent the oil from splashing on the cylinder walls and piston. As it was impracticable to prevent some of the oil from splashing, it was decided to reduce the oil pressure and thus reduce the amount of oil splashed. The pumps had previously been carrying an oil pressure of 14 pounds per square inch (above atmosphere), whereas by design 5 pounds per square inch was sufficient for proper lubrication. But as is usual with bearing trouble, the policy had been to "give her more oil" and the pressure had been increased with only more disastrous results. So the average oil pump pressure was re-

duced to 7 pounds, and the machinists on watch had orders not to let it go over 8 pounds. In addition, the cylinder lubrication from the mechanical oilers which had been running about 18 to 20 drops per minute was cut to 4 drops per minute. In addition, after each long run, the oil passages were thoroughly cleaned out, especially those in the crankshaft webs. A small amount of the tarry mass was discovered to have collected each time, but not sufficient to cause trouble. With this treatment, practically all bearing trouble disappeared for a while. It reappeared again, however, on a trans-Atlantic trip of the O-7. The cause appeared mysterious at first and it was feared that something was wrong with the above theory. But the new trouble was soon located on investigation. It was found that the lubricating pump was frequently getting air bound and losing its suction, resulting in the engine more or less frequently running for short intervals without any lubrication. This resulted in the main bearings wiping sufficiently to plug the oil holes and subsequent running burned out the bearings. Nearly all bearings on one engine were burned out at sea from this cause on the return trip. The pump was found to be getting air bound from air leaks in the system on the suction side of the pump. Some trouble was experienced in definitely locating all these air leaks, but when they were all found and remedied the bearing trouble ceased to exist.

Some vessels of this class had piston seizures which were probably caused primarily by the same cause as above. The wrist-pin bearings would heat up, being short of oil from clogging in the system and the heat would be transferred to the piston walls. This would cause excessive expansion of the piston and cause it to stick and seize in the cylinder. The oil residue left on the cylinder walls from its partial burning would be a contributory cause of this result. The treatment as above to eliminate bearing trouble also ended the piston seizures.

This information sums up as follows:

(*a*) Burned out main and crank-pin bearings; piston seizures.

(1) Caused by too high lubricating oil pressure and subsequent excessive splash on hot piston and cylinder walls, resulting in gummy residue mixing with lubricating oil.

(*b*) Remedies.

(1) Reduce lubricating oil pressure to slight margin above theoretical requirements, viz., 7 pounds.

(2) Reduce cylinder wall lubrication to minimum, about 4 drops per minute.

(3) Clean oil passages to all bearings and in hollow crank-shaft after each long run.

(4) Prevent air leaks on suction side of pump, so that pump will not get air bound and lose suction.

(5) Keep bearings adjusted to proper clearance.

ECONOMY OF LUBRICATING OIL.*

USE OF LUBRICATING OIL AS FUEL.

The following procedure has been tried out on the U. S. S. O-7 but is not advised as good practice for usual routine unless in case of emergency.

The O-7 on a trip across the Atlantic to the Azores was running a very high consumption of lubricating oil, giving grave concern over whether the supply would last for the trip. As it was not possible to stop to go over the system to discover whether the bearings had the proper clearances and whether or not this was the cause of the high consumption, as

* In the opinion of the Editor the writer was saved from very disastrous results by prevailing conditions; the practice outlined has nothing to recommend it, but rather should serve as a bad example. Discussion of this article is invited.

no external leaks to the bilge could be found, and as it was imperative to reduce the consumption, the experiment was tried of mixing fresh water with the oil so that half of the expenditure would be water instead of oil. This worked out very satisfactorily so far as could be discovered on a 14-day run. The fresh water, about 10 to 15 gallons, was simply added to the settling tank every four hours. This amount would churn up with the oil and form a good emulsion. Once every twenty-four hours sufficient fresh oil would be added to bring the level in the settling tank to the proper point. By the use of fresh water, the amount of oil needed to be added each day was cut about in half. The mixture of oil and fresh water formed an emulsion which was a sufficient lubricant and the water was evaporated off upon striking the hot parts of the system, thus reducing the amount of lubricating oil burned. The only loss of lubricating oil, it must be understood, in a system similar to this one is either from leaks to bilge or burning of the oil on the cylinders and other hot parts of the engine. By allowing the fresh water to be "burned" or evaporated in place of oil, considerable saving resulted. Great care must be observed, however, to see that fresh water and not salt is used as salt water will leave a heavy salty deposit which will ruin the engines if continued. From using fresh water as above, however, for over 10 days' continuous running, no harmful results were observed, and a big saving of lubricating oil was accomplished.

The use of lubricating oil as fuel came about as a result of the above economy in lubricating oil. Several days before the end of the 14-day trip it was foreseen that fuel oil was going to be very short at the end of the run and everything possible was done to economize. But the evaporator had been run almost constantly to furnish fresh water for the crew for bathing and washing as well as for diluting the lubricating oil, which all resulted in increased fuel consumption. It is also believed that the heavy consumption was par-

tially caused by extra power used in driving line shafts which had been sprung and propellers which had been damaged by striking part of the superstructure which had been previously carried away by heavy seas. Anyway, every drop of fuel was used when the O-7 was about 12 hours out from Punta Delgada, Azores. Due to the economy in *lubricating oil,* there was a surplus left of about 400 gallons. To burn it in the engines as fuel was the problem, as the engines were designed to burn a lighter oil, something between boiler fuel and the lighter distillates. It was first attempted by mixing 10 per cent of alcohol (which was kept in the vessel for use in torpedoes) with lubricating oil. It was feared that the alcohol might not mix thoroughly and that if the cylinders received a quantity of pure alcohol, a disastrous explosion might result. However, the mixture was made in buckets and thoroughly stirred and then poured into the gravity feed tanks. After a few splutterings and misfires, the engines started off and appeared to run all right. It worked first rate and filled us with joy that we would be able to make port without assistance, although it kept the crew busy mixing alcohol and lubricating oil and pouring it into the tank. All went well for about 4 hours, when the alcohol gave out and the destination about 8 hours away. Attempt was made to burn the lubricating oil without alcohol, but it would not seem to fire. However, after many attempts and with the storage battery getting low from frequent attempts to start the engine, it was found that if the engine circulating water was choked down enough to allow the cylinders to warm up to a temperature of about 160 degrees F. the lubricating oil would burn all right.

The reason for this is probably two-fold. It is believed that the rise in temperature of circulating water resulted in somewhat increasing the temperature of compression, and also that the mixing chamber of the spray valve became hotter from the increase in temperature of the circulating water

and that this resulted in better flow of the lubricating oil through the spray nozzles and consequently a finer spray. Anyway, following this procedure good combustion was obtained and for about 8 hours no further trouble burning straight lubricating oil for fuel was experienced. The vessel reached the destination with both engines running and a supply of lubricating oil sufficient to run for quite a number of hours more.

However, the above practice is uneconomical and probably results in excessive carbonization of cylinders, imperfect combustion, and might result in burning spray valve nozzles and in damage to fuel oil pump. So it is not advocated as standard practice unless in an emergency to reach port with all fuel exhausted. The method used in burning it, however, is a good thing to know in case an emergency demands its use.

The mixing of water with lubricating oil for lubricating purposes is not advocated as standard practice as it is not believed to be necessary under proper conditions and might result in corrosion of journals or bearings if carried out for a long time. The possibility of its use, however, is a good thing to bear in mind if ever caught out at sea with lubricating oil running low and no other way to reach port.

TEST OF TWO-TON CLOTHEL REFRIGERATING MACHINE.

BY M. C. STUART, ASSOCIATE.

There was conducted recently at the U. S. Naval Engineering Experiment Station, Annapolis, Md., a test of a two-ton refrigerating machine manufactured by the Clothel Co., New York City. Inasmuch as Refrigerating Machines of this type in smaller sizes are in use in the Naval Service, it is believed that a description of the two-ton machine and a brief summary of the results obtained on the test would be of interest.

The refrigerant used in the machine is Ethyl-Chloride, C_2H_5Cl, a neutral chemical having a boiling point of 54.5 degrees F. at atmospheric pressure. The unit consists of a motor-driven rotary compressor, a condenser, separator, expansion trap and brine cooler. The motor-driven compressor is shown in Fig. 1. The motor drives the compressor through spur gears having a speed reduction of 3.75 to 1. Small rotary pumps for circulating the brine and the condensing water are driven from either side of a spur gear which is driven in turn by the large gear. The construction of the rotary compressor is shown in detail in Fig. 2. The rotor is located eccentrically in the cylinder which is bored elliptically on a special machine. The top of the rotor makes contact with the top of the cylinder, thereby forming a seal between the suction and pressure sides of the cylinder. The blades are fitted into four slots milled radially in the rotor and into the outer edge of the blades are fitted half round packing strips which are machined to conform to the inside of the cylinder. The blades and packing strips are held out by spacing pins which pass through the center of the shaft. The seal between the blade and cylinder wall is therefore positive and contact between blade and cylinder wall does not depend

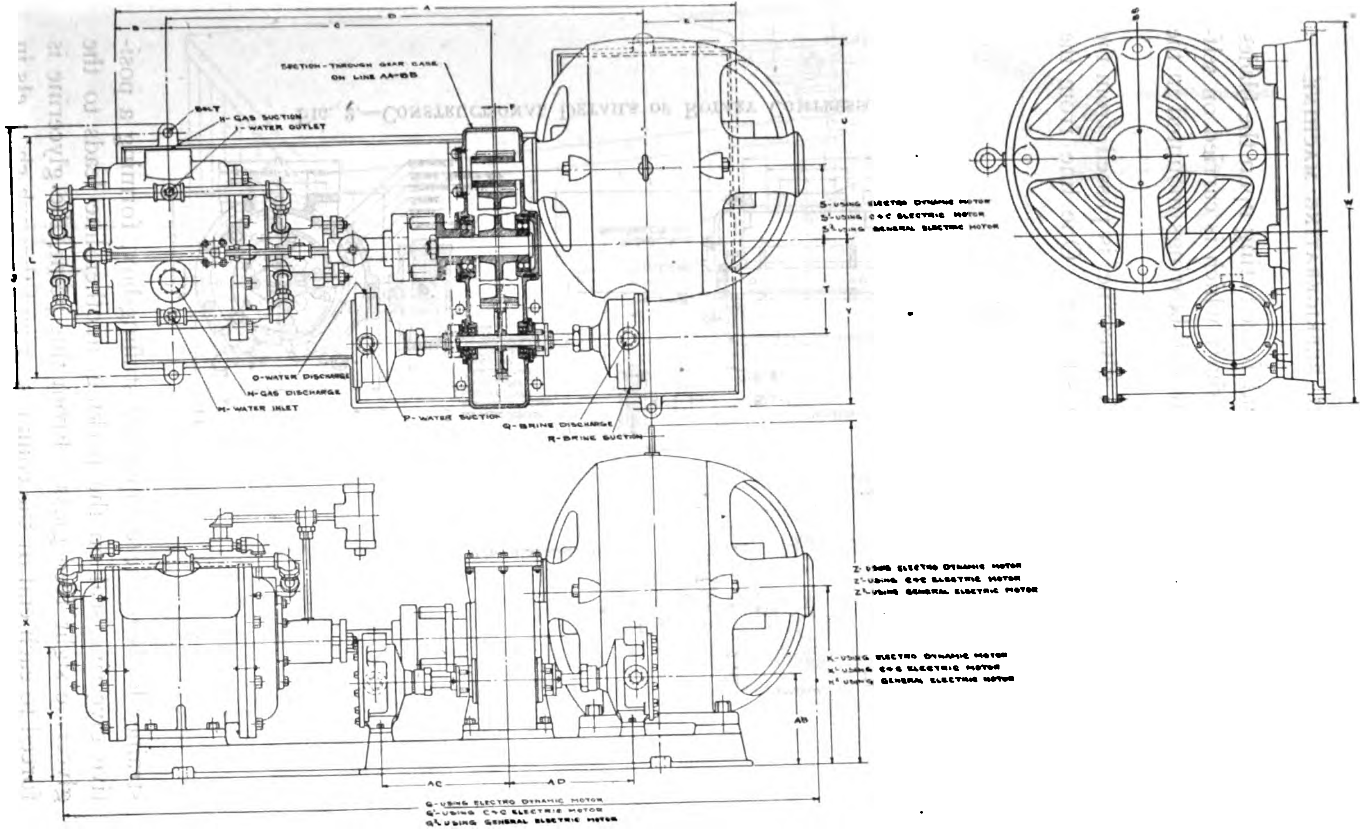


Fig. 1.—Motor-Driven Rotary Compressor.

upon springs or upon centrifugal force acting upon the blades. The rotor is keyed to the steel shaft which is carried on self-aligned, heavy duty, double-row ball-bearings mounted in the cylinder heads.

The lubricant, which is C. P. glycerine, is passed from the condenser through a glycerine strainer. One line from the

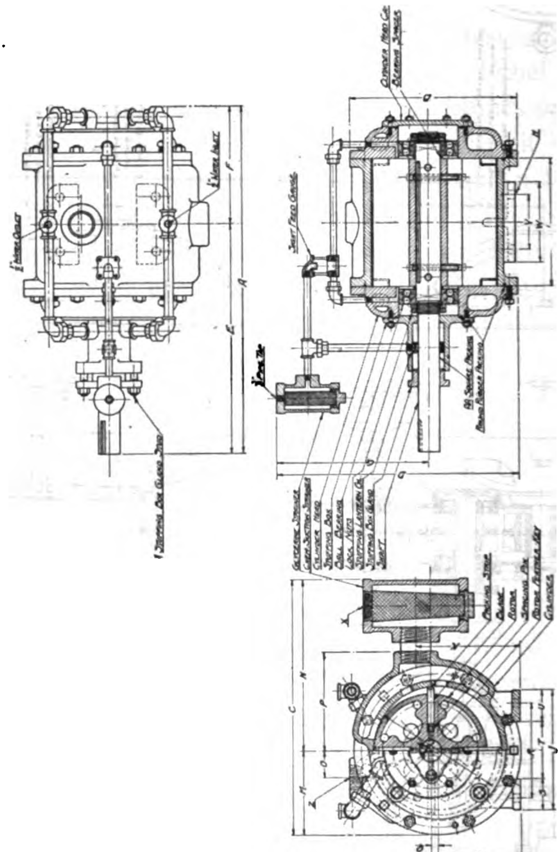


FIG. 2.—CONSTRUCTIONAL DETAILS OF ROTARY COMPRESSOR.

strainer leads to the compressor stuffing-box, forming a positive glycerine seal on the packing, another line leads to the glycerine sight feed glass. From this point, the glycerine is forced to each end of the compressor and through channels in

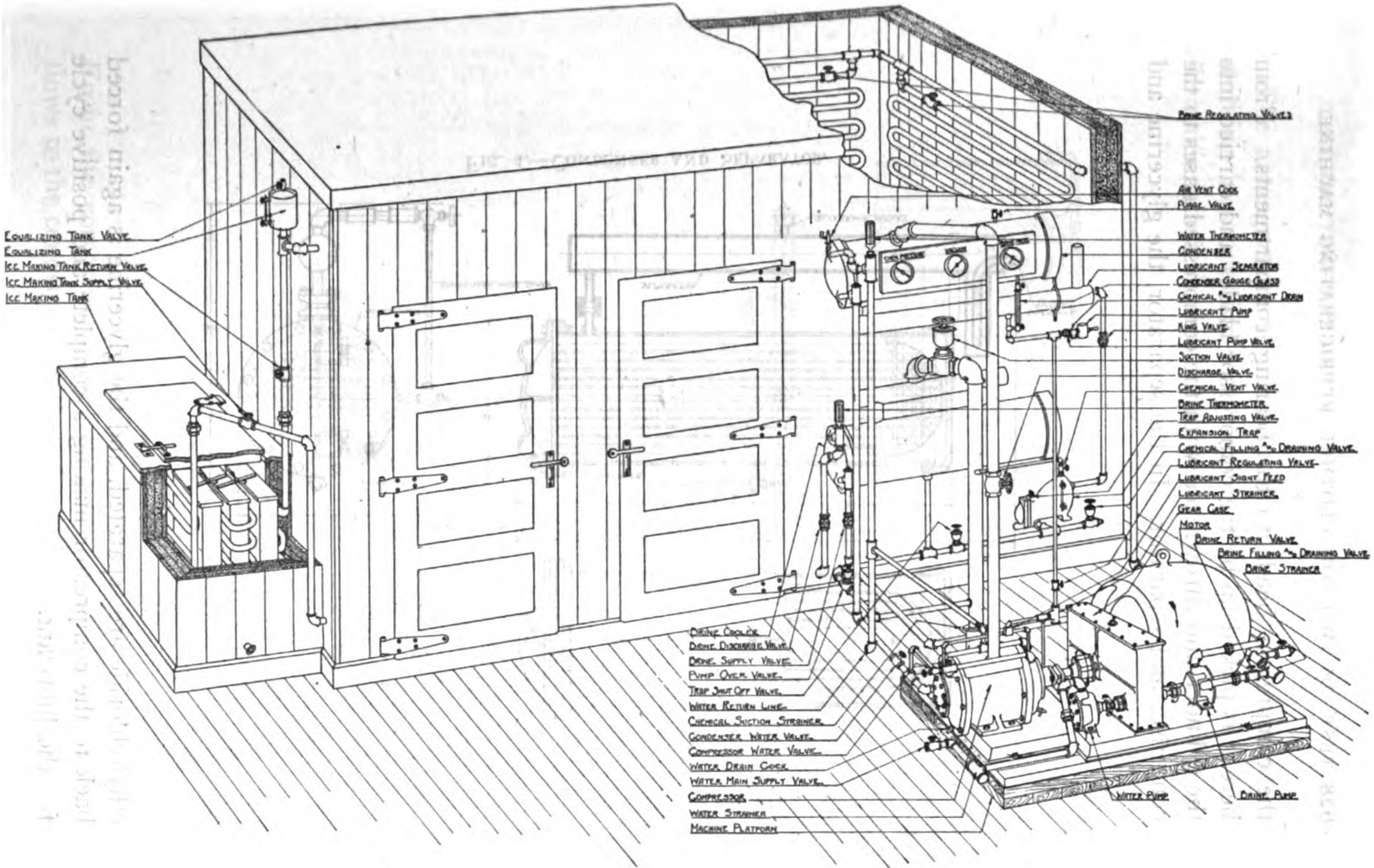


FIG. 3.—PERSPECTIVE OF COMPLETE TWO-TON CLOTHEL UNIT.

the compressor heads to the bearing compartments. From here it is picked up by the compressor blades and carried into the cylinder and thrown out with the compressed gases to the condenser and separator. In the separator, the glycerine and

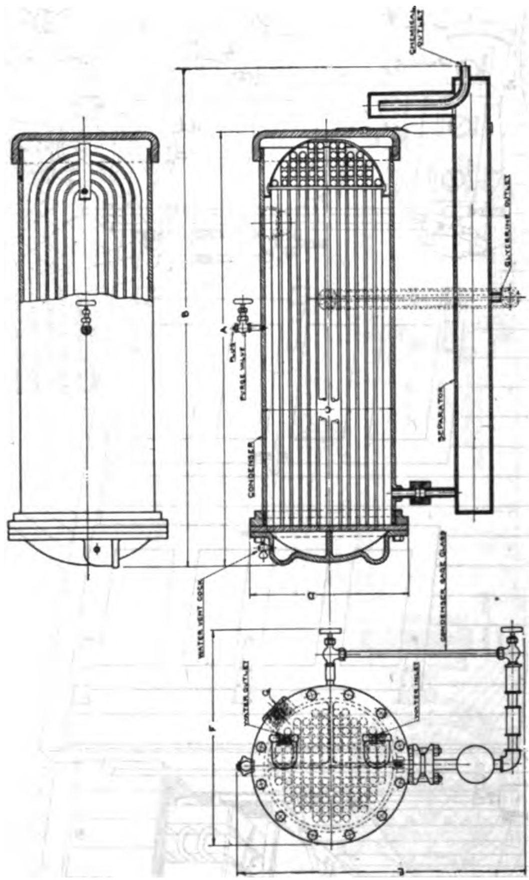


FIG. 4.—CONDENSER AND SEPARATOR.

ethyl-chloride are separated, and the glycerine is again forced back to the compressor making a complete and positive cycle for the lubricant.

The arrangement of the complete refrigerating apparatus working in connection with a cold-storage compartment and ice-making tank is shown in the isometric drawing, Fig. 3.

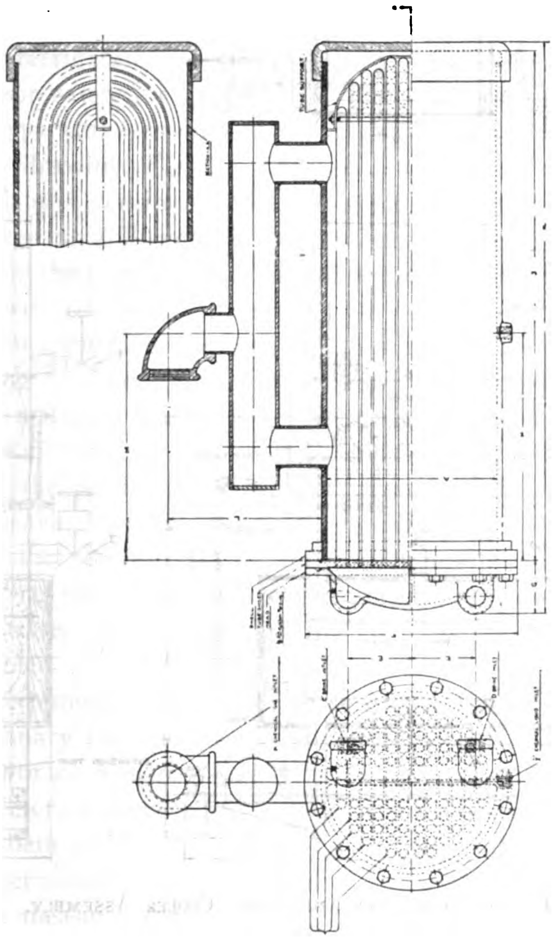


Fig. 5.—Brine Cooler.

From the compressor, the refrigerant in the form of gas flows to the condenser which is shown in detail in Fig. 4. The

condensing element consists of U-tubes arranged in two passes with condensing water flowing through the tubes. After being condensed, the liquid refrigerant, together with

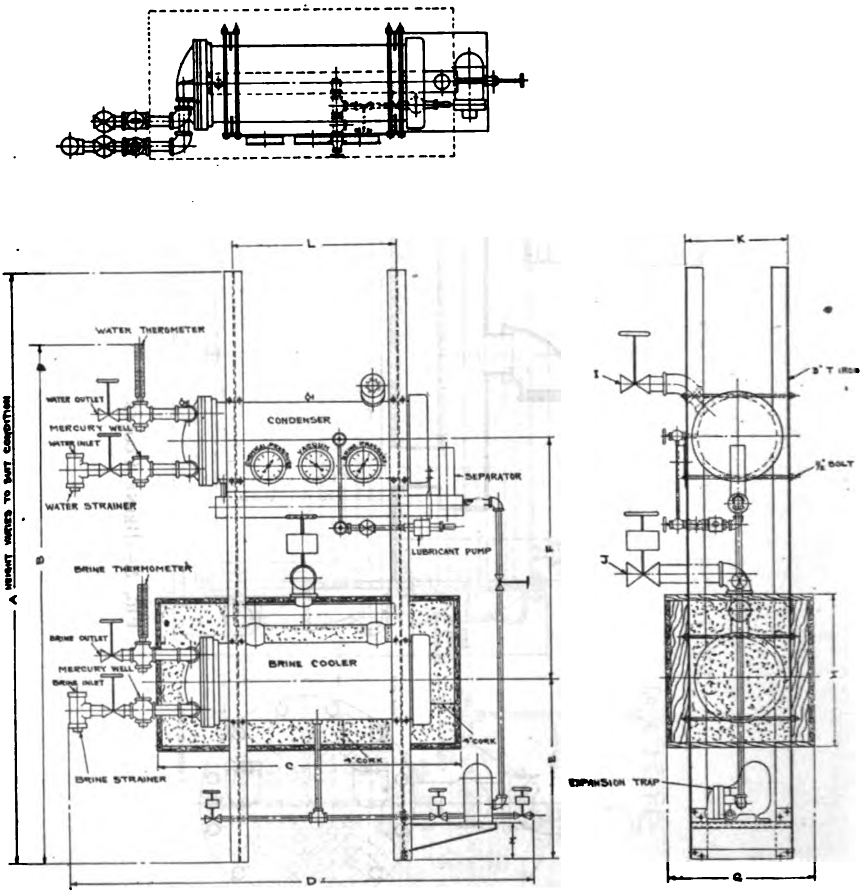


FIG. 6.—CONDENSER AND BRINE COOLER ASSEMBLY.

the lubricant, falls into the separator where the lubricant, being heavier than the refrigerant, is drained back to the compressor and the liquid refrigerant is lead to the expansion trap as shown in Fig. 3. After expansion through the valve

of the expansion trap, the refrigerant passes to the brine cooler which is shown in detail in Fig. 5. The construction of the brine cooler is similar to that of the condenser. The refrigerant is in the shell and the brine passes through U-tubes arranged in two passes. In the brine cooler, the refrigerant is vaporized by the heat which is abstracted from the brine, and the refrigerant, in the form of a vapor, passes to the compressor where the cycle of operations is repeated.

The apparatus was erected for test exactly as shown in the assembly drawing, Fig. 3. The speed of the motor was determined by a continuous reading speed counter. Pressures were taken as follows: of the refrigerant leaving the compressor, in the condenser, in the brine cooler, and entering the compressor; of brine entering the brine pump, leaving the brine pump, entering the cooler and leaving the cooler; of the circulating water entering the condenser and leaving the condenser; entering the compressor water jacket and leaving the compressor water jacket. Temperatures were taken as follows: of the refrigerant leaving the compressor, in the condenser, leaving the separator, leaving trap in the cooler, and entering the compressor; of the condensing water entering and leaving the condenser and entering and leaving compressor water jacket; of the brine entering and leaving the brine cooler.

All the temperatures except those of the brine were taken with ordinary mercury thermometers reading to 1-degree in which 10th of a degree could be estimated. This type of thermometer, however, was not sufficiently accurate for the measurement of brine temperatures, because the measurement of refrigeration in the method used during the test depended upon the measurement of the drop in temperature of the brine passing through the cooler, and inasmuch as this drop in temperature would usually amount to only two or three degrees, thermometers in which 10ths of a degree must be estimated were not sufficiently accurate. For the purpose of

measuring inlet and outlet brine temperature, special thermometers were used having graduations to 10ths of a degree and in which 100ths of a degree could be estimated. The thermometers were read by means of telescopes and by interchanging the inlet and outlet thermometers several times during each run any possible error in the differential temperature between the inlet and outlet brine could be determined and eliminated. From the analysis of the brine temperature readings thus obtained, the average possible error in temperature difference was found to be less than 0.02 degree F.

The electrical input to the motor was measured by means of an ammeter in the armature circuit, an ammeter in the field circuit, a voltmeter connected across the armature, and a voltmeter connected across series and interpole fields. From these electrical readings, the output of the motor was determined by means of the stray power method.

Condenser circulating water and the cylinder jacket circulating water were weighed in tanks on platform scales. The amount of brine circulating through the brine cooler was measured by means of a calibrated Venturi meter which was located in the line between the brine pump and the brine cooler. This Venturi meter was calibrated by comparison with weighings of brine on calibrated scales. During the run the brine from the brine cooler was circulated through a brine heater, where by means of circulating water sufficient heat was added to bring the brine to the temperature desired for the run. From the brine heater the brine was discharged into a storage tank from which the brine pump took its suction. By this means a very constant brine temperature of any desired value could be obtained. The reading of a differential mercury manometer on the Venturi meter indicated the quantity of brine flow. During the calibration of the Venturi meter, the brine was discharged into tanks on scales mounted above the brine storage tank, and conditions of operation were kept the same as during the runs.

Runs were made with circulating water temperatures of approximately 65, 70, and 90 degrees F. With 65 degrees circulating water, 4 runs were made with outlet brine temperature of approximately zero, 10, 15 and 20 degrees F. With 70 degrees circulating water, runs were made with outlet brine temperatures of zero, 10, 15 and 20 degrees. With 90 degrees circulating water, runs were made with outlet brine temperatures of zero, 10, 15, 20, 30 and 35 degrees F. All runs were made at the rated speed of the machine. Runs were not started until the conditions became constant and each run was of a duration of one hour.

RESULTS OF TEST.

The principal results obtained on the test are given in Table I. The total refrigerating effect was determined from the heat given up by the brine in the brine cooler. In order to obtain the net available refrigerating effect, the amount of heat added to the brine by the brine circulating pump was subtracted from the total refrigerating effect. The heat given up by the brine in the brine cooler was computed from the product of quantity of brine circulated, drop in temperature of brine in passing through cooler, and specific heat of the brine. The value of specific heat of the brine was obtained from the Bureau of Standards Scientific Paper No. 135.

An inspection of the table of results shows that the refrigerating effect is increased with higher brine temperatures and lower circulating water temperatures. With a circulating water temperature of 70.1 degrees F., and a brine temperature of 14.57 degrees F. (which is practically the ice-making temperature) the net refrigerating effect was 2.13 tons per day.

TABLE I.

Item No.	Item	Unit														
1	Run number.	–	1	2	3	4	5	6	7	8	9	10	11	12	13	14
2	Speed of motor.	R. P. M.	837	849	807	815	809	831	836	840	833	843	842	846	846	844
3	Speed of compressor.	R. P. M.	223	226	215	217	215	221	222	223	222	224	224	225	225	225
4	Temperature of circulating water.	Deg. F.	89.7	89.9	90.0	90.0	89.9	89.7	70.1	64.1	64.9	65.2	64.5	70.5	70.1	70.3
5	Temperature of brine outlet.	Deg. F.	(–)0.15	10.00	14.63	20.19	30.14	35.05	9.66	(–)0.01	15.21	9.69	19.90	20.16	14.57	(–)0.05
6	Power to motor.	H. P.	9.01	10.05	8.85	9.35	9.86	10.01	7.57	7.25	8.03	7.39	7.95	8.36	8.22	7.90
7	Vapor pressure in condenser.	Lbs. per sq. in. gage.	33.2	37.7	33.3	35.7	38.8	37.8	21.0	17.0	21.1	18.9	21.3	24.9	23.6	22.9
8	Temperature of Ethyl-Chloride in condenser.	Deg. F.	98.2	100.8	103.8	105.8	109.0	112.7	84.1	76.2	83.0	79.5	83.1	88.3	86.7	83.2
9	Vacuum in brine cooler.	Inches mercury.	23.4	21.7	20.8	19.7	17.3	15.9	22.0	23.6	20.8	21.8	19.7	19.6	20.9	23.6
10	Temperature of refrigerant in brine cooler.	Deg. F.	(–)8.5	(–)1.0	3.0	7.0	16.0	19.9	(–)2.0	(–)10.0	2.7	(–)1.9	7.0	7.0	3.0	(–)9.6
11	Brine circulated.	Lbs. per min.	213.2	213.7	202.1	202.3	198 8	204.0	208.0	211.3	206.5	213.0	210.4	211.7	209.4	211.7
12	Condensing water.	Lbs. per min.	109	108	100	104	98	91	100	100	98	100	99	98	95	92
13	Temperature drop of brine in cooler.	Deg. F.	1.75	2.30	2.94	3.11	3.79	4.26	2.72	2.18	3.14	2.70	3.38	3.32	3.06	2.10
14	Refrigerating effect, total.	Tons per day.	1.25	1.66	2.02	2.15	2.58	3.00	1.92	1.55	2.21	1.95	2.43	2.41	2.19	1.49
15	Refrigerating effect, net.	Tons per day, net.	1.18	1.59	1.96	2.09	2.52	2.94	1.86	1.48	2.14	1.88	2.36	2.34	2.13	1.42

CHARLES H. MANNING.

In the death, April 1, 1919, of Chief Engineer Charles H. Manning, U. S. Navy, retired, there passed away an officer whom the Navy of his day honored for his engineering genius, his sterling character, and his kindly and genial qualities which won men's friendship everywhere. Loyal, gallant, faithful in all things—thus he lived to honored age. And, in passing, he leaves noble memories to the engineering Navy of our day.

Charles Henry Manning was born in Baltimore, Md., on June 9, 1844. He was the son of Joseph Cogswell Manning and Rebecca Parkman Jarvis (Livermore) Manning, and came from old New England stock, having been the descendant in the ninth generation of William Manning, who settled in Cambridge, Mass., in 1634, and whose land there remained in the Manning family for more than two hundred and thirty years thereafter. In 1871, Captain Manning married Miss Fanny Bartlett, of Boston, the sister of Major-General William F. Bartlett, one of the most gallant men sent by Massachusetts to the Civil War.

Mannings' father was for many years a prominent iron merchant of Baltimore, owning and operating the Avalon Nail and Iron Works there. He received his early education in private schools in Baltimore and in the High School of Cambridge, Mass. In 1860, he entered the Lawrence Scientific School of Harvard University to study civil engineering. After the fall of Fort Sumter, he joined the drill club which guarded the State Arsenal at Cambridge, and later was offered a lieutenancy in a Massachusetts regiment, which, because of his youth, his father would not permit him to accept.

Owing to his father's business reverses brought on by the war, Manning returned to Baltimore in the autumn of 1861, and, following his natural bent, entered as an apprentice the marine engine works of Charles Reeder where, in working on

naval vessels, he met a number of their engineer officers. Contact with them gave his active energies a definite aim, and, as a result, he was appointed a Third Assistant Engineer in the Navy on February 19, 1863, having passed third in the class of sixteen of that date.

He was, however, to see relatively little service under fire, since Engineer-in-Chief Isherwood, recognizing his worth as a scientific observer, kept him closely occupied with the classic experiments on superheated steam in the *Adelaide* and other vessels, thus limiting his experience of actual naval war to some brief fighting in Hampton Roads and to passage in convoy of troops during the siege of Charleston.

He served on the *Adelaide* for two years, and left her in March, 1865, for the sloop-of-war *Dacotah* to which vessel he was attached until September, 1868, making in her the circuit of South America and cruising as far north as Panama. The outward passage was noteworthy for the company of the Peruvian ironclads *Independencia* and *Huascar* from Rio de Janeiro onward, since the latter vessel was to become memorable in naval history for her heroic fight in the subsequent war with Chile. The *Dacotah* was also at Valparaiso during the great earthquake of August, 1868, and the next day left for Africa, where 10,000 lives had been lost and the U. S. S. *Wateree*—an iron double ender—had been washed a mile inland and left there on an even keel.

From the *Dacotah,* Manning was ordered to the sloop-of-war *Seminole* which cruised on the home station, mainly in West Indian waters, until February, 1870, when she returned to New York with yellow fever on board, and was put out of commission. From September, 1875, to September, 1877, he served on the sloop-of-war *Swatara,* and from September, 1880, to August, 1882, on the President's Yacht, *Despatch*—in both cases on the home station. With the latter duty, his service at sea ended.

Owing to his marked ability, not only in engineering science but as an instructor, Manning's shore duty was passed wholly

at the Naval Academy, where he was stationed for five years beginning with September, 1870; and later for three years from September, 1877, onward. During the first of these tours of duty, he aided in organizing the course of instruction for Cadet Engineers at the Academy—in fact, as the subsequent experience of his students proved, his influence had a most marked and lasting effect on that course.

"This was the most valuable work I ever did," said the veteran engineer to the writer one evening long ago, as he reviewed the achievements of a life, strenuous always, and successful in so many fields. And, he might well say this, for the courses which he had so vital a part in founding constituted not only the first definite effort by any navy to train young men for its Engineer Corps, but formed as well the first technical school of mechanical engineering established in the United States. The work of Manning and his associates—Kafer, Tower, Kearney, and others—in this aided signally in the advancement of the sciences of marine and mechanical engineering.

The engineering course at Annapolis was established by Congress in 1864, but the project languished until 1871 when the first class of sixteen members was admitted to the Academy. Yearly thereafter, twenty-five cadets were appointed until 1882, when, in the full tide of its success, it was abolished by Act of Congress, and the Cadet Engineers then at the Academy were transferred as Cadet Midshipmen to the Line of the Navy.

The course met thus its sudden ending because of its very success, and that success was the result primarily of two conditions: first, its instructors were picked men, the ablest in the Engineer Corps of that day; second, its cadets were selected by competitive examinations, so severe eventually that most of the successful candidates came from colleges or universities.

The natural result of this system was the formation of a body of students who, in scientific ability, could not but be superior, as a whole, to the Cadet Midshipmen of that day,

who were then, as a rule, appointed by political influence. The contrast between the two courses was sharply shown when, in 1882, the Cadet Engineers were transferred to the corresponding classes of Cadet Midshipmen. In his work, "The Steam Navy of the United States," Captain Frank M. Bennett, U. S. Navy, says:

"A year after the exchange was made, we find that seven of the first ten members of the graduating class were former Cadet Engineers, and four of the six 'stars' of that class were engineers. * * * In the next class, the three leading members were former Cadet Engineers. * * * In the third class, after the amalgamation took place, * * * we find that thirteen of the first fifteen members, nine of the first ten, and all six of the 'stars' were former Cadet Engineers."

All navies were then, like our own, in a transition stage, passing slowly and reluctantly from sails to steam. The inevitable and sweeping progress of engineering in naval science was then far from being foreseen by even engineers themselves. It is not strange, then, that the great body of Line Officers, proud of the noble traditions of the days of sail, should have dreaded the effect of such a system of engineering education upon the prestige of their branch of the service, and so the change was made. Even that accomplished seaman, Captain "Jack" Philip, when asked why the course had been abolished, said frankly: "We had to do it. If we had not, the engineers would have crowded us off our ships." From the viewpoints of naval efficiency and of harmony within the Navy, it is regrettable that the effective solution of this vital problem—given by the Amalgamation Act of 1899—was not sooner found, and that engineering was not earlier made, as it should be always, one of the duties of the Line of the Navy.

In 1881, while attached to the *Despatch,* Manning served as a member of the first Advisory Board. This Board consisted of thirteen officers of the Line, Engineer Corps, and Construction Corps. The engineer members, in addition to Manning,

were Chief Engineers Benjamin F. Isherwood and Charles H. Loring. In one noteworthy step of the Board's action—the decision to build the hulls of the new vessels of steel—Manning cast the deciding vote, the Board standing 7 to 6, and he being the only engineer who, in the existing condition of steel-making in the United States, was willing to take the responsibility of voting for that metal.

Manning's service to his Corps, in seeking by all just means the rightful recognition of engineering, began early as was to be expected from his determined character. The Line and Staff feud of those old days was a relentless strife, the bitterness of which it is difficult to realize fully at this time. Two incidents, which Manning used to tell with a grim smile, show his boldness in forcing justice.

A bill had been rushed through, without due consideration, in the closing days of Congress, abolishing the grade of Third Assistant Engineer. If this measure had become a law, its effect would have been to drop summarily these officers from the navy, wherever they might be stationed, "from China to Peru," without even an allowance for traveling expenses to their homes. By unremitting effort, Manning, through a friend, brought the bill to the indignant attention of President Andrew Johnson on the last day of the latter's term of office. The President said: "This measure will become a law today, with or without my signature. There is but one thing to do," and, with his own hands, he tore the bill in two and threw the fragments into a fire blazing in an open grate in his office at the White House. With Congress on the verge of adjournment and the only legal copy of the bill thus destroyed, the threatened officers were saved to the Navy.

Shortly after, Manning was informed by the chairman of the Naval Committee of the Senate that that body would confirm appointments of certain engineer officers if their nominations were sent in at once by the Navy Department, but that a further delay would be dangerous. He called upon the Secretary of the Navy immediately and stated the case. The

Secretary, new to his office, sent for Admiral Porter, who, at that time, was virtually supreme in naval affairs. The Admiral, eager to defeat the nominations, advised against them; but Manning, in his presence, presented such a forcible argument to the Secretary that, despite the Admiral's opposition, the nominations were sent in and promptly confirmed. Only those familiar with naval customs can understand fully the boldness of a young officer of twenty-five in thus antagonizing the Admiral of the Navy. Such temerity could have, in those days, but one result—a prompt cruise to the torrid West Indies, to which Manning was shortly ordered.

It is pleasant to note that Admiral Porter, while a relentless enemy with an unfading memory, had a manly respect for a worthy foeman. Years after, recognizing Manning on the *Despatch,* he came to the engineer officer and entered into a long and pleasant conversation, as if there never had been war between them.

In May, 1884, Manning was placed on the retired list of the Navy, owing to a partial loss of hearing, due, as the medical board reported, to exposure " in the line of duty." The retirement of such an officer, otherwise in the full vigor of manhood, was a most serious loss to his Corps and the Navy, a view which Chief Engineer Isherwood expressed in an appreciative letter to him, and which Rear-Admiral John L. Worden, who had commanded the *Monitor* in her engagements with the *Merrimac,* and who was then the president of the Retiring Board, also held, since he said to Manning: " I would rather see any other man than you in your Corps go."

In August, 1882, Manning was granted a year's leave of absence after twelve years of continuous duty. This leave was given that he might become Mechanical Engineer of the Amoskeag Manufacturing Company, of Manchester, N. H. Subsequently, he became General Superintendent of this company, remaining as such until 1913 when he retired and opened offices as a Consulting Engineer.

The Amoskeag Company operated the largest cotton mills in the world, its land and buildings extending in an almost unbroken line on both sides of the Merrimac River for a mile and a half. As superintendent, Captain Manning had charge of the power plant and the buildings and grounds, and in addition was the architect and builder of new mills there, including the latest, the Coolidge Mill, erected during recent years. The output of these mills is about a million yards of cloth a day. Under his administration, the steam power plant grew from one of 870 square feet of grate surface burning about 400 tons of coal per week, to one of 5,800 square feet, consuming 3,000 tons.

These mills show throughout the traces of Captain Manning's ability as an engineer and organizer. In the beginning of his service, he re-assembled the boiler plant after a plan striking in its boldness. The widely scattered units were gathered into groups near the coal pile, a steam pipe was laid across the river, and, in one case, steam was led through a pipe nearly half a mile long. Then, he designed the Manning vertical water-tube boiler, which is used not only in the Amoskeag Mills but very largely in textile manufacturing in New England.

In 1885, he was an engineering pioneer in designing and building at the Amoskeag Mills the first large installation (2,000 horsepower) of water turbines on a horizontal shaft. The fall of water was 47 feet and critics claimed that the sidewise thrust on the wheels would affect their operation. They, however, ran smoothly from the beginning. In 1891, a flywheel, 30 feet in diameter and running at 60 revolutions per minute, burst in one of these mills, the cause being a defective casting. Manning replaced it with his historic 30-ft. "wooden flywheel," an iron wheel with a rim built almost wholly of wood. The rim has a face of 108¼ inches and a thickness of 12 inches, and was made up of forty rings of ash.

Manning's service during the Spanish-American War proved the sterling quality of his patriotism. It was known to all

well-informed officers that the naval fighting in that war could be but brief and relatively unimportant, since our force on the sea was overwhelmingly superior to that of Spain. Manning was then fifty-four years of age, his business responsibilities were great, and he had the strongest political and naval influence, since both the Secretary of the Navy and the Engineers-in-Chief were his personal friends.

In these conditions, many retired officers of his age would have been tempted to seek assignments to relatively light inspection duty near their homes. Not so with Manning. There was important work to be done at the Key West Naval Station, in the repair of the machinery of the many warships which gathered there. Owing to the torrid climate, there was no more unpleasant shore duty for a naval officer. But, when Manning was asked to go there as Chief Engineer, he went without question and spent the summer in the stifling heat, glad to serve the flag again.

Years ago, in correspondence with the writer, Rear Admiral Melville, formerly Engineer-in-Chief of the Navy, said of Manning's work there:

"How fast the men of the old Engineer Corps are fading! There are left now but few of that bright galaxy—Kafer, Manning, Thurston, West, and a half score of others—who established the course for cadet engineers. To my mind, the ablest and most level-headed of them all was Manning. He was big in every way—in build and brain and heart.

"During the Spanish-American War the Navy Department was disturbed by the unusual number of ships which were laid up for repairs at Key West. For various reasons the situation at that station was difficult to handle. As engineer-in-chief, I tried several engineer officers there, giving them an abundance of tools and material, but the ships still lingered. Finally, I sent Manning there and the atmosphere of delay cleared quickly. He put no extra polish or finish on the work, but all that tended to efficiency of the machinery was done, and done promptly, and, through his tact and energy, the ships were

soon despatched to their stations on the southern side of the island of Cuba."

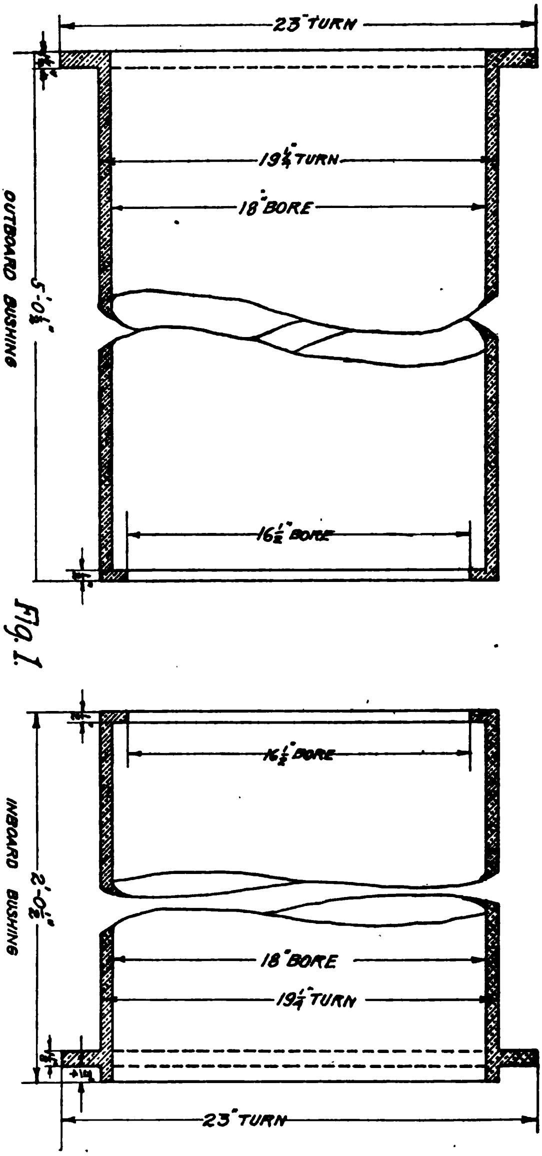
Captain Manning was active in civic duties. He was a delegate to the New Hampshire Constitutional Convention of 1886, and served for many years on the Engineering Committee of the Board of Visitors at Harvard University. In Manchester, he was a member of the School Board for twenty years or more, and for many years a trustee of the Elliot hospital. As an authority on water rights and hydraulic engineering generally, he was for twenty-eight years a member of the Board of Water Commissioners—most of the time its president—and was, by far, the most influential figure in the development of the city's notably good water supply.

He was a Past Vice-Commander of the Military Order of the Loyal Legion, and a Past Vice-President and Honorary Member of the American Society of Mechanical Engineers. In 1895, Harvard University conferred on him the degree of Bachelor of Science, as of his old class of 1862. He was a member of the American Society of Naval Engineers, the United States Naval Institute, the American Society of Naval Architects and Marine Engineers, the American Association for the Advancement of Science, and the American Society of Cotton Manufacturers.

In closing this brief sketch, the writer feels that it records but inadequately the forceful and brilliant career of the gallant officer who was his instructor and friend in years gone by. Every inch a man, and in every fibre an engineer, Captain Manning, in his long service, not only honored his profession and the Navy, but his manly and generous nature won for him as well a return which, when all is said, is better still—the deep respect and the warm friendship of many men.

PROBLEM CORNER.

Solutions of previous problems.



Problem No. 1. See Fig. 1.

Problem No. 2. No solution submitted.

Problem No. 3. The moment of the circular segment about the y axis, see Fig. 2, is

$$M = 2\int_{R\cos\theta}^{R} xy \; dx = 2\int_{R\cos\theta}^{R} x\sqrt{R^2 - x^2} \; dx = \tfrac{2}{3} R^3 \sin^3\theta.$$

This moment M is maximum when $\frac{dM}{d\theta} = 0$ or when

$\frac{d}{d\theta}(\frac{2}{3} R^3 - \sin^3\theta) = 2 R^3 \sin^2\theta \cos\theta = 0$, this occurs when $\sin\theta = 1$ or $\theta = 90°$. The counterbalance of maximum effect is therefore a semicircle.

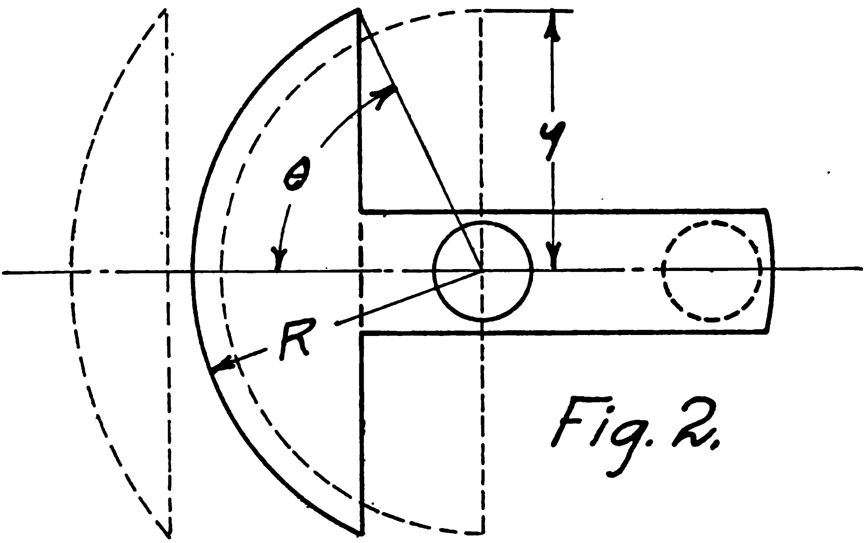


Fig. 2.

Since $M = \frac{2}{3} R^3 \sin^3\theta$ and $\sin\theta = \frac{y}{R}$, $M = \frac{2}{3} y^3$, which is to say, all segments having the same center and the same breadth have the same moment.

From a practical standpoint the semicircular counterbalance is too heavy and interferes with machining when forged on. The thickness of the crank webs determines the thickness of the counterbalance. The thickness of the

counterbalance and its maximum radius being determined, its breadth and distance from the axis can be computed from the expression for M.

Problem No. 4.

If a rod be suspended from one of its points the time of a single beat is $t = \pi \sqrt{\frac{k^2 + h^2}{g\,h}}$, k being the radius of gyration about the center of gravity and h the distance of the point of support from the center of gravity.

Suspended from one end $t = \pi \sqrt{\frac{\frac{1}{12} l^2 + \frac{1}{4} l^2}{\frac{1}{2} g\,l}} =$ $\pi \sqrt{\frac{2}{3} \frac{l}{g}} = 1$ if it is to make 60 single beats per minute,

hence $1 = \pi^2 . \frac{2}{3} \frac{l}{g}$ and $l = \frac{3 \times 32.2}{2 \times 9.8696} = 4.9$ feet.

For another point of suspension to give the same period, h must have such a value that

$$t = \pi \sqrt{\frac{\frac{1}{12} l^2 + h^2}{g\,h}} = 1, \text{ or } \frac{\frac{1}{12} l^2 + h^2}{g\,h} = \frac{2}{3} \frac{l}{g} \text{ hence}$$

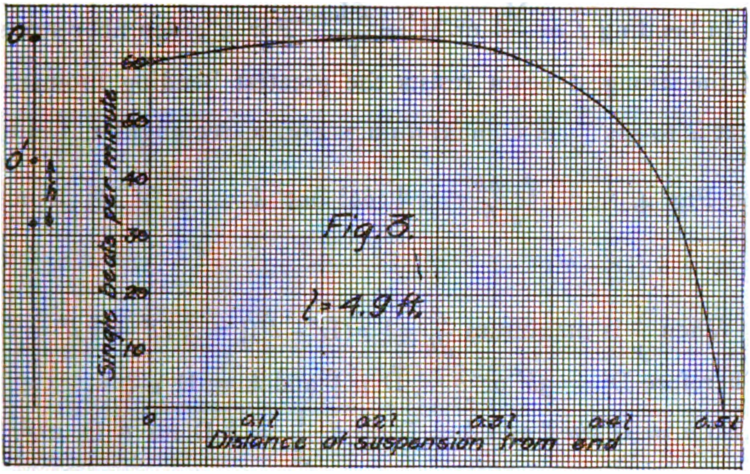
$$3\,h^2 - 2\,lh + \tfrac{1}{4}\,l^2 = 0 \text{ and } h = \tfrac{1}{2}\,l \text{ or } \tfrac{1}{6}\,l, \text{ therefore}$$

$$oo' = \tfrac{1}{3}\,l = 4.9 \div 3 = 1.63 \text{ feet.}$$

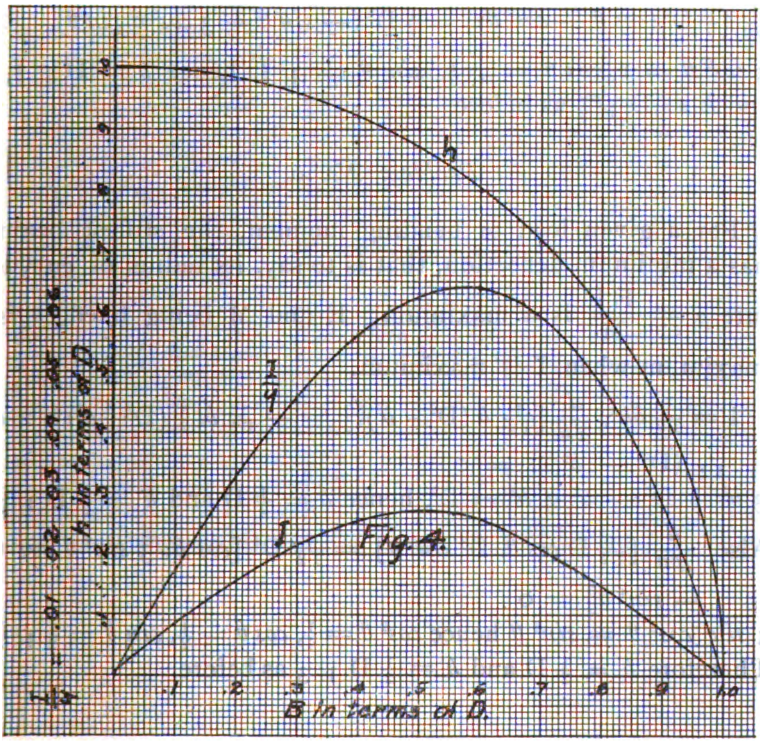
The maximum rate occurs when t is a minimum, that is, when $\frac{\frac{1}{12} l^2 + h^2}{h}$ is minimum or when $\frac{d}{dh}\left(\frac{\frac{1}{12} l^2 + h^2}{h}\right) =$ $-\frac{l^2}{12} . \frac{1}{h^2} + 1 = 0$, then $h = \frac{l}{\sqrt{12}} = \frac{4.9}{3.464} = 1.41$ ft.

the corresponding rate is $\frac{1}{t} = \frac{1}{\pi \sqrt{\frac{\frac{1}{12} l^2 + \frac{1}{12} l^2}{g \frac{l}{\sqrt{12}}}}}$

or $\frac{1}{t^2} = \frac{6g}{\pi^2\,l\,\sqrt{12}}$ or $\frac{1}{t} = 1.08$ beats per second or 64.8 beats per minute. See Fig. 3.


O
O'
Single beats per minute
60
50
40
30
20
10
Fig. 3.
l = 4.9 ft.
0
0.1l
0.2l
0.3l
0.4l
0.5l
Distance of suspension from end

Problem No. 5.


h
Fig. 4.
h in terms of D
B in terms of D.
.1
.2
.3
.4
.5
.6
.7
.8
.9
1.0

In bending $f = \frac{M\bar{y}}{I}$ or $M = \frac{fI}{y}$. For a given fiber stress f, the strongest beam resists the greatest bending moment, that is to say $\frac{I}{y}$ is a maximum. For a rectangular beam, see Fig. 4, $\frac{I}{y} = \frac{\frac{1}{12}\,bh^3}{\frac{1}{2}\,h} = \frac{1}{6}\,bh^2$ but $h = \sqrt{D^2 - b^2}$, hence $\frac{I}{y} = \frac{1}{6}\,b\,(D^2 - b^2)$.

When $\frac{I}{y}$ is maximum $\frac{d}{dh}\left[\frac{1}{6}\,b\,(D^2 - b^2)\right] = \frac{1}{6}\,D^2 - \frac{3}{6}\,b^2 = 0$.
Therefore $D^2 = 3b^2$ or $b = D\sqrt{\frac{1}{3}}$ and $h = D\sqrt{\frac{2}{3}}$
Also $h = D \sin\theta$ and $b = D \cos\theta$, $\frac{I}{y} = \frac{1}{6}\,D \cos\theta\, D \sin^2\theta$ $= \frac{1}{6}\,D^2 (\cos\theta - \cos^3\theta)$. When $\frac{I}{y}$ is maximum
$\frac{d}{d\theta}\left[\frac{1}{6}\,D^3 (\cos\theta - \cos^3\theta)\right] = \frac{1}{6}\,D^3 (-\sin\theta + 3\cos^2\theta \sin\theta) = 0$
$3\cos^2\theta = 1$, $\cos\theta = \sqrt{\frac{1}{3}}$, $\sin\theta = \sqrt{\frac{2}{3}}$, hence $b = D\sqrt{\frac{1}{3}}$ and $h = D\sqrt{\frac{2}{3}}$ as before.

Problem No. 6.

For a given load or bending moment, the stiffest beam has a minimum deflection, that is to say, its radius of curvature $\rho = \frac{EI}{M}$ is a maximum; E and M being fixed, ρ is maximum when I is maximum.

$I = \frac{1}{12}\,bh^3$, $\frac{d}{dh}\left[\frac{1}{12}\,b\,(D^2 - b^2)^{\frac{3}{2}}\right] = -\frac{1}{12}\,b.\,\frac{3}{2}\,(D^2 - b^2)^{\frac{1}{2}}.\,2\,b$ $+ \frac{1}{12}\,(D^2 - b^2)^{\frac{3}{2}} = 0$
$3\,b^2 = D^2 - b^2$, $4\,b^2 = D^2$, hence $b = \frac{1}{2}\,D$ and $h = \frac{1}{2}\,D\sqrt{3}$
Also $\frac{d}{d\theta}\left[\frac{1}{12}\,D\cos\theta.\,D^3\sin^3\theta\right] = \frac{1}{12}\,D^4\left[\cos\theta.\,3\sin^2\theta\cos\theta - \sin^4\theta\right] = 0$
$3\sin^2\cos^2\theta = \sin^4\theta$ hence $\cos^2\theta = \frac{1}{4}$, $\cos\theta = \frac{1}{2}$, $\sin\theta = \frac{1}{2}\sqrt{3}$
Therefore $b = \frac{1}{2}\,D$ and $h = \frac{1}{2}\,D\sqrt{3}$ as before.

Problems Nos. 7 and 8.

See Fig. 5.

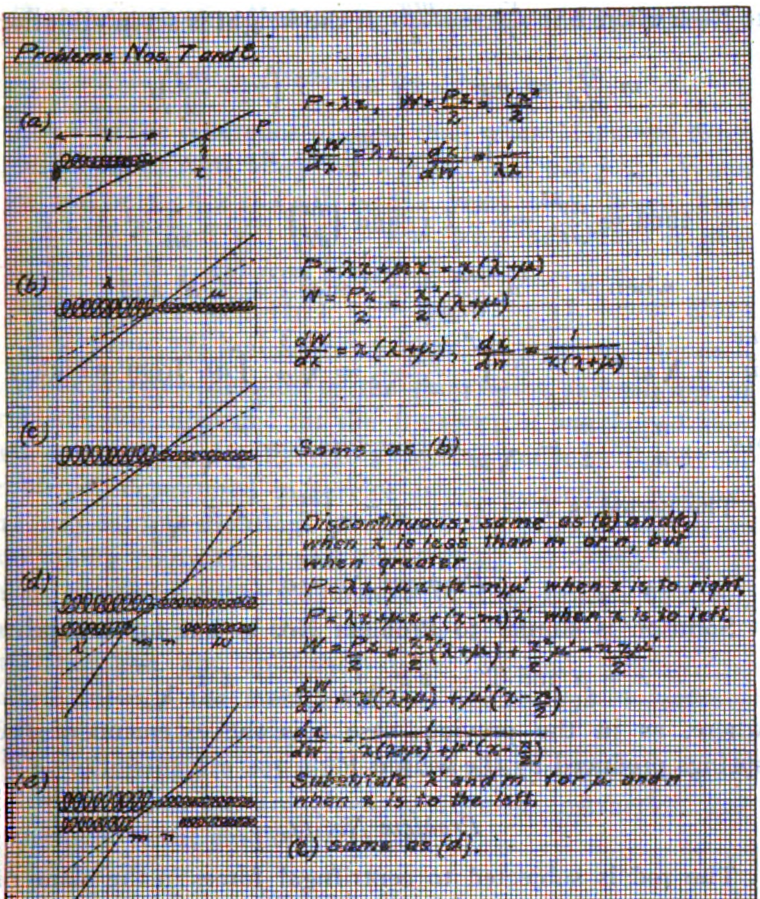


Fig. 5.

Problem No. 9.

A weight of P pounds lengthens the supporting spring to $l + e$ where $e = \frac{P}{\lambda} = \frac{mg}{\lambda}$. If the weight is set in vibration

$$m \frac{dy^2}{dt^2} = mg - \lambda y = -\lambda \left(y - \frac{mg}{\lambda}\right),$$

$$\text{let } y - \frac{mg}{\lambda} = x$$

$$\text{then } m \frac{dy^2}{dt^2} = -\lambda x, \text{ and } \left(\frac{dx}{dt}\right)^2 = -\frac{\lambda}{m} x^2 + c$$

$$\text{when } x = a, \frac{dx}{dt} = 0, \text{ hence } c = \frac{\lambda}{m} a^2 \text{ and}$$

$$\frac{dx}{dt} = \sqrt{\frac{\lambda}{m}(a^2 - x^2)}, \int_0^a \frac{dx}{\sqrt{a^2 - x^2}} = \sqrt{\frac{\lambda}{m}} \int_0^t dt, \text{ from which}$$

$$\sin^{-1} \frac{a}{a} = \frac{\pi}{2} = \sqrt{\frac{\lambda}{m}}\, t, \text{ therefore } t = \frac{\pi}{2} \sqrt{\frac{m}{\lambda}}$$

The period is $T = 4t = 2\pi \cdot \sqrt{\frac{m}{\lambda}} = 2\pi \sqrt{\frac{e}{g}}.$

Problem No. 10.

Solution is similar to Problem No. 9 but for λ appears

$$\lambda + \mu \text{ and } T = 2\pi \sqrt{\frac{m}{\lambda + \mu}} = 2\pi \sqrt{\frac{e}{g}}$$

where, as in the previous case e is the extension of the springs produced by the weight P.

Problem No. 11.

The rod is subjected to centrifugal forces in the plane of the axis and to accelerating forces in a plane perpendicular to the axis until the motor comes up to speed. Hence the rod is subject to greatest stress when both forces are acting and when both are greatest, this occurs just before uniform speed is reached.

The work done in accelerating the armature and rod from zero to n R. P. M. is $W = \frac{1}{2} \omega^2 \; (I_a + I_\rho)$.

Since this is done uniformly in t seconds, the power expended is $P = \frac{1}{2}(I_a + I_\rho)\frac{4\pi^2 n^2}{3600} \times \frac{1}{550t}$.

The system is out of balance both statically and dynamically.

Problem No. 12.

Let x be the difference in ages, then when Mary was three times as old as Ann, Mary was $\frac{3}{2}x$ and Ann was $\frac{1}{2}x$. When Ann will be $3 \times \frac{3}{2}x$ she will be $\frac{9}{2}x$ years old. When Mary was $\frac{1}{2} \times \frac{9}{2}x$, Ann was $\frac{9}{4}x - x = \frac{5}{4}x$, then Mary is now $2 \times \frac{5}{4}x = \frac{10}{4}x$ and Ann is now $\frac{10}{4}x - x = \frac{6}{4}x$.

But $\frac{10}{4}x + \frac{6}{4}x = 44$, $4x = 44$, $x = 11$, hence Mary is $\frac{10}{4} \times 11 = 27\frac{1}{2}$ and Ann is $\frac{6}{4} \times 11 = 16\frac{1}{2}$.

When a space devoted to the proposition and solution of problems was suggested for the JOURNAL, it was hoped that such space might develop into a clearing house for the actual problems encountered in service ashore and afloat, research and original investigation. Research and original investigation, along engineering and Naval lines are of the utmost value, as pointed out by Commander C. S. McDowell, U. S. N., in the June number of the *Proceedings of the Naval Institute.* Practical, live, and up-to-the-minute problems can be furnished only by the members of the Society and the readers of the JOURNAL. *Come across* !

The following are not sufficiently serious to deserve a place in the JOURNAL but are given to furnish instruction and amusement to the studiously inclined.

Problem No. 13.

A ship A maintains speed and course. Another ship B, m miles abeam of A overtakes A by maintaining speed and changing course so as to always head for A. If A steams a knots and B steams b knots, how long does it take B to overtake A, how far does each travel and what is the course of B?

Problem No. 14.

If, as we carelessly put it, force times distance is work and force times lever arm is torque, how can it be explained that the dimension of both products is the same?

Problem No. 15.

A closed cylindrical vessel, filled with water, is made to revolve about its axis. In calculating the pressure at various points it is common to apply the expression for centrifugal force $\frac{mv^2}{r}$. Show that this is altogether wrong and work out the correct formula for two cases, (*a*) when the height of the cylinder is negligible and (*b*) when it is considerable.

Problem No. 16.

If a projectile weighs 125 pounds, how much would it weigh on the moon? How much difference would there be in the recoil in firing it?

Problem No. 17.

A heavy triangular boiler plate, sides *a*, *b*, and *c*, is lying on the floor. Which corner is the lightest to lift (opposite side resting on the floor) and which is the lightest to carry the plate by?

Problem No. 18.

A pile of coal is on deck, more is coming on board through a chute. Eleven men can clear the deck in twenty minutes, six men can clear it in sixty minutes, how long will it take four men?

NOTES.

BRITISH SUBMARINE BOAT BUILDING DURING THE WAR.

(Abridged.)

Messrs. Vickers' achievement in submarine construction during the war is one of which the firm has every reason to be proud. Fifty-four submarines of all classes have been built and commissioned in a period of fifty-one months of war, and are detailed as follows:

"E" class	15
"V" class	4
"G" class	6
"K" class	6
"N" class	1
"H" class	10
"L" class	9
Other types	3
Total	54

Of these boats, all except "V" and "N" classes were built to Admiralty design.

These figures are additionally interesting when it is remembered that in practically every class Messrs. Vickers were the pioneer builders. The outbreak of the war found the company in a remarkably good position for the rapid output of submarines. The type which was then being perfected, and of which several were in commission, was the now well-known "E" class.

This "E" class is one of the most numerous in the fleet. These vessels, as their performances have shown, proved themselves suitable for many types of work. Two of them, A.E. 1 and A.E. 2, after completion at Barrow, proceeded under their own power to Australia and took up duties with the Australian Navy. One of these vessels actually had considerably over 30,000 miles to her credit before it was necessary for her to undergo a refit to her propelling machinery. As sea-going boats they were a distinct step in advancement on previous classes. The extensive superstructure combined with the navigating bridge built over the conning tower made them easy to navigate in even the roughest weather. This advantage is the more appreciated when it is understood that only in the finest weather could a "B" or "C" class boat cruise on the surface with open hatches.

The overall length of the "E" class is 181 feet, beam 22 feet 7 inches over the side tanks, and the displacement when submerged is 780 tons. As regards armament, five 18-inch diameter torpedo tubes are carried, two in the bow, two on the broadside, and one in the stern. Each tube is provided with one spare torpedo, thus making a total of 10 torpedoes carried. The vessels have been fitted also with guns of various types, but a tabulated statement would be necessary to give full particulars of these.

The radius of action at a cruising speed of 10 knots is 3,225 nautical miles, with a total oil fuel capacity of 42 tons. A surface speed of 15 knots is obtained by twin screws which are each driven by a Vickers' eight-cylinder heavy oil engine of four-cycle, vertical, single-acting type, each engine developing 800 brake horsepower. The motors, for use when submerged, are of the Vickers' single-armature open type, each developing 420 brake horsepower, and resulting in a speed of 10 knots. The power for the motors is stored in two separate batteries of 112 cells each, constructed by the Chloride Electrical Storage Company, of Manchester. Air storage is provided by 57 bottles stowed about the ship. High-pressure air compressors keep the bottles charged.

To equip these boats for more effective scouting and patrol work, a wireless telegraphy installation is fitted, the aerial being carried on folding masts. In addition, a set of under-water signalling gear is carried. Two periscopes are fitted, one being a simple, single-power instrument, the other being fitted with a device which readily permits of alternative powers of 1 and 6 being used. The latter can also be used as a "sky-searcher" for detecting enemy aircraft.

Messrs. Vickers created a record as regards the building and equipment of the "E" class submarines, E. 19 being built, equipped and handed over to the British Admiralty in exactly eight months; whilst a flotilla of six, including the above, was commissioned in thirteen months from the date of verbal Admiralty instructions to proceed with their construction. During the period in which the company were engaged on "E" class construction, they were also devoting considerable attention to a new type of submarine in which the principal idea was the provision of a complete double hull.

The "G" class followed upon the Vickers' "B" design, to which reference will be made later. This is of increased size and armament. The "G" class (of which Messrs. Vickers constructed six) is 187 feet long, 22 feet 8 inches beam, with a submerged displacement of 840 tons. These dimensions follow the single-hull "E" class very closely, but with increased displacement consequent on the addition of the double hull. The full surface speed of these vessels is 15½ knots, obtained by twin Vickers' heavy oil engines, each developing 800 brake horsepower. The full speed submerged is 10 knots, the motors working at 420 brake horsepower each. The armament of the "G" class consists of two bow and two broadside torpedo tubes, 18-inch diameter, and one stern tube, 21-inch diameter, with a spare torpedo for each tube. The guns are two in number, situated on the superstructure at either end of the conning tower casing, one being of the disappearing type 3-inch Q.F., and the other 2-pounder H.A.

To augment the submarine fleet as quickly as possible, in the early months of the war, Messrs. Vickers placed the resources of their Canadian establishment at Montreal at the disposal of the British Admiralty. As a result of this arrangement 10 submarines of the "H" class were added to the service. These were designed by the Electric Boat Company, of America, and were sister vessels to some of the later submarines in the United States Navy. They formed a very handy type of small submarine, and did useful work in the Adriatic as well as in the North Sea. Their seaworthiness was fully proved during the voyage across the Atlantic. In December, 1916, the Admiralty placed an order with Messrs. Vickers for 12 boats of the "H" class, modified slightly from the American design to accommodate heavier torpedo equipment, and wireless telegraphy gear of a much higher power. Their length overall is 171 feet 9 inches, beam 15 feet 9 inches, and submerged displacement 510 tons. They have a speed of 13 knots on the surface, and 10 knots submerged. Twin screws are fitted, and the engines develop 480 brake horsepower. The electric motors have a total normal shaft horsepower of 640.

The chief feature in this class is the very powerful torpedo equipment in relation to their size. Four 21-inch torpedo tubes are arranged in the bows, and six torpedoes are carried. As in the case of previous classes Messrs. Vickers were entrusted with the first boat of the "H" class, the experience of their designing and constructive staffs making them eminently suited for pioneer work of this kind.

A class of "J" boats were built in the dockyards, and had each three sets of Diesel engines by Messrs. Vickers. These were the fastest heavy oil-engined vessels in the world.

The "L" class combines in design the best qualities of other boats, but with increased speed. These three vessels were delivered simultaneously—an incident which brought to the firm a special note of congratulation from the Admiralty.

The circular single hull is used with the side tanks as in the "E" boats, but the stern is a modified and improved arrangement of the stern fitted in another class. The length overall is 238 feet 6 inches, being 57 feet longer than the "E" class. The beam over the side tanks is 23 feet 6 inches, and the total submerged displacement is 1,090 tons. The Diesel machinery is of a more powerful type than previously fitted. The twin screws are worked by four-cycle, 12-cylinder oil engines and develop a total of 2,600 brake horsepower, which results in a surface speed of over 17 knots. Motors for submerged running are fitted to both shafts, and on a total brake horsepower of 1,600 a speed of 10½ knots can be maintained. The auxiliary machinery includes two high-pressure air compressors of the inverted water-cooled type, each capable of delivering per hour 22.5 cubic feet of air compressed to 2,500 pounds per square inch.

The torpedo equipment in this class consists of four 21-inch tubes in the bows, and two 18-inch tubes in the broadside room, with a spare torpedo for each of the tubes. A 4-inch gun is mounted on the superstructure just forward of the bridge and conning tower. Very large angles of training are obtained by the use of a revolving platform which moves with the gun. A novel feature is the night periscope which is fitted in addition to the ordinary instruments. This enables the boat's crew to keep a more effective watch at night.

As the submarine requirements of the Allied Grand Fleet increased, it became necessary to have a type which, whilst still retaining the ordinary functions of the submarine, could yet maintain its station with the fleet when cruising. These requirements were embodied in the design of the "K" class, of which Messrs. Vickers were entrusted with the first ship of the type, and also five others. These vessels have a displacement when submerged of 2,570 tons, being 339 feet long overall, with a beam of 26 feet 8 inches. The double hull principle is embodied in a modified form, and the ends of the boat which might be damaged, without endangering the immediate safety of the boat have only a single hull at the sides. Their speed is 24 knots on the surface, the power being obtained from twin sets of geared steam turbines, which develop a shaft horsepower of 10,500. Steam is obtained from two boilers of the Yarrow type, with a working pressure of 235 pounds per square inch. The funnels are arranged to hinge down when preparing for diving; water-tight hatches cover the funnel openings as the funnels hinge back. The turbine machinery is supplemented by an 800 brake horsepower heavy oil engine of the Vickers submarine type, which is coupled up to a dynamo of the open single armature design. This arrangement enables the turbines to be reserved for higher speeds only, whilst the dynamo, in addition to charging batteries, supplies the main motors with power for cruising at economical speeds. For submerged work the motors develop a total horsepower of 700 per shaft, being arranged in tandem on each shaft, and give a speed of 9 knots. The motors

drive the shafts through helical gearing. The storage battery for the use of the motors is divided into three groups of 112 cells per group.

Owing to the very exacting duties required of these boats, additional protection for the navigating officer and quartermaster is provided in the form of a deckhouse built over and around the ordinary conning tower. The periscopes fitted into these boats were the longest which at that time had been made for the British Navy, having a length of 30 feet from eye-piece to head-piece. Another important accessory is the long-distance wireless telegraphy installation, enabling the commander of the submarine to keep in touch with his naval base when operating at great distances from it. The aerial is carried on telescopic masts, which can be raised and lowered at will. The air compressing machinery consists of two low-pressure and two high-pressure air compressors, with air bottles stowed in various positions about the vessel contiguous to their work. A telemotor system is installed for raising the periscopes and telescopic masts, and also for working the ballast tank vent valves.

The armament of this class includes four bow and four broadside tubes, all 18-inch diameter, with a spare torpedo for each tube. The guns are carried on the superstructure and consist of two 4-inch guns and one 3-inch gun for high-angle work.

These vessels are a departure from the ordinary type in that they are self-contained units of the fleet, and do not rely upon a parent ship as is the case of "E" and other classes. Their performances during the war have been of the greatest value to the naval authorities, their speed, armament and low visibility making them exceedingly useful additions to the British Navy. It is interesting to note that this class of submarine is, for surface work, the fastest in the world.

The "V" class was introduced at an earlier date as a result of tank experiments by Messrs. Vickers in which several forms of hull were tried. Four vessels of the "V" class were ordered by the Admiralty from the firm. The principal dimensions are length overall 147 feet 6 inches, beam of outer hull 16 feet 3 inches, and submerged displacement of 460 tons. Twin sets of Vickers heavy oil engines are fitted, developing 450 brake horsepower each, from which a surface speed of about 13 knots is obtained, the engines being of the four-cycle vertical single-acting type with eight cylinders. When submerged a normal full speed of 8½ knots is obtained from two motors each developing 190 brake horsepower. The storage battery consists of 132 cells arranged in two groups of 66 cells each. The torpedo equipment of these boats consists of two 18-inch diameter bow tubes and two spare torpedoes.

Minelaying submarines are a logical outcome of the war, and several of the "E" and "L" classes were equipped as such. The mines are loaded into vertical tubes which pass through the external tanks. Mechanical gear is fitted for both locking the mines in position and for releasing them when required.

The Vickers' engine being the standard for the British boats, the existence of spare-gear stocks all over the world prevents any radical departure in the design of the principal parts, but nevertheless, in developing the 8-cylinder reversing design, it was possible to make the main parts interchangeable with the 12-cylinder non-reversing type, and at the same time to make the weight less than that of the non-reversing engine of the same power and cylinder dimensions.

The construction of both engines is similar, so that in this respect one description suffices. The cast steel main bearing girders are connected longitudinally by forged beams. There are plate columns between the cylinder covers and the main bearing girders to relieve the cylinders of all vertical stress. (These engines are shown on folders in the next article.)

A cast-iron liner is dropped into a sandwich plate between the covers and columns, and is thus free to expand downward. A jacket of steel plate, corrugated for expansion, is fixed to the sandwich plate at the top, and is fitted with a sliding joint at the bottom between the jacket and the liner. Thrust reactions are taken by snugs butting against the bottom of the liner where it passes through the top plate of the crank-case. This top plate serves to tie the columns together.

The cylinder covers are of cast steel of simple construction, secured to the sandwich plate and to the columns. The joint is made by a cast-iron ring ground into grooves in the liner and in the cover. Water connections between the jacket and cover consist of tubes through the sandwich plate, passing into holes in the bottom of the cover, water-tightness being secured by soft packing rings around the tubes. No submarine engine cover has been known to fail in a Vickers' engine.

The valve boxes are separate from the cover, and are fitted upon coned seats in recesses in the cover. The valve boxes are water cooled by water from the cover. This water, in the case of the exhaust valve, passes into the jacket of the exhaust bend. The valves are of nickel steel, and the exhaust valve is water-cooled through flexible hoses. The piston is of cast-iron, with six upper rings and one wiper ring. They are non-water-cooled. An aluminum guard is fitted above the gudgeon to prevent oil splashing upon the crown. The connecting rod is hollow, of round section, with separate bottom end and compression liner. The crank is closed by portable sheet casings, and is ventilated by suction pipes to the induction header, non-return valves being fitted in the suction pipes to eliminate any possibility of a backfire into the crank-pit. The above-named features are common to both engines.

We come now to points of difference between the two engines. The 12-cylinder engine has the inlet and exhaust cams on a lower camshaft actuating the valves through push rods. The upper camshaft carries the fuel cams and drives the fuel injection pumps, one to each cylinder. This separate pump design is that originally preferred by the Service, but the spray valves have proved themselves such accurate means of regulating the fuel injected that a common fuel supply is now finding favor, and this is fitted in the reversing design. In this latter case the fuel is pumped into a common main from which are branches leading to the separate cylinders. No gags are required, as for air injection engines with a common fuel supply. This common supply pump is a four-throw pump driven by spirals from the end of the crankshaft. In the 12-cylinder engine the pump discharges may be connected at will, thus resulting in running on the "common rail" system as used in the eight-cylinder engine. The use of the common pump permits of one camshaft only in the reversing design. This shaft carries all cams, the air-starting cams being at the ends of the engine. The short push rods for the inlet and exhaust rods are pulled away from the cams by the rotation of a fulcrum shaft, after which the camshaft is slid longitudinally, and the push rods are then replaced upon the cams for running in the opposite direction. The fuel and air cams are bevelled, permitting the camshaft to slide without their tappets being lifted. All reversing is carried out by hand, it being desirable to avoid servo-motor gear, which generally gives trouble after some time at sea.

The 12-cylinder engine has the standard spray valve with external gland. The reversing engine has a flooded valve, in which the spindle does not project through a gland, but is actuated by a bell crank, the rotating shaft passing through a gland in the side of the valve-box. Both types give equally good results, and choice is made according to the ease of adaptation to the valve mechanism.

In each case the engine is started by air. The air is admitted in correct sequence by tappet valves with a spring and a balance piston upon them. By this means the valves are lifted off the cams when the air master valve is closed, and they only come into action when starting or reversing is required. The air passes to each cylinder through a non-return valve on the cover. The mechanical injection is so safe in its operation that there is no necessity to shut off air before admitting fuel. In the reversing engine it is arranged that the starting wheel shuts off air when all cylinders are on fuel, but in the non-reversing type the two are not interconnected. No cylinder relief valves are necessary for these engines, although in the reversing engines such are fitted to comply with the standard specification. The fact that the British submarine engine has no cylinder relief valves is practical evidence of the strength of the parts and of the safety of the injection system.

The control is different in the two engines. We may describe first that of the 12-cylinder engine. One hand wheel in the center controls the opening of all of the spray valves. Individual cylinders can be put in or out of action by the handles at each cylinder. A second hand wheel controls the output of the fuel pumps by sliding a shaft upon which stirrups moving the scroll cams actuating the suction valve of each pump are mounted. The horizontal wheel advances or retards the injection by moving vertically a coupling in the vertical shaft. This coupling is free to slide vertically on one part of the shaft, but drives the other through inclined keys. By this gear the upper camshaft can be rotated a few degrees relatively to the crankshaft. This gear is a refinement, enabling the spray cams to be set with great convenience to meet any conditions of fuel or load-speed ratio, which may vary considerably in a submarine charging when running on the surface, and it is not fitted to the reversing engine in which the duration of injection is so related to the first instant of injection throughout the range of control that automatic regulation sufficient for satisfactory running in all ordinary conditions is obtained. An emergency stopping valve is also fitted to the 12-cylinder engine, which releases the pressure from the rail.

The eight-cylinder engine control may now be described. The large hand wheel (Fig. 3 of next article) is for reversing, and a small hand wheel interlocked with the first wheel first admits air to all cylinders, then fuel to four cylinders, then fuel to the second four, and shuts off air. A further turn stops the engine by shutting off fuel. The pump output is controlled by a small lever regulating the closing point of the suction valves. The large upper lever controls the spray valve opening.

All cylinders in both engines are 14½ inches diameter by 15-inch stroke, and each easily develops 100 brake horsepower at 380 R.P.M. Powers up to 50 per cent in excess of this have been obtained at higher speeds, but a conservative rating has been followed by the Admiralty authorities. It is plain that high ratings must result in reduced durability, and the consistently steady running of the British submarines year in and out appears to point to the soundness of this policy.

The injection is of the Vickers' mechanical system, no injection-air compressor with its weight, space, complication and danger being required. Air compressors, however perfect the design, are considered the most troublesome parts of Diesel engines, and in high duty engines the supply and control of the high-pressure air is a serious problem. The system consists simply of a high-pressure pump supplying the fuel to a main, from which a definite amount is admitted each firing stroke by a measuring valve in the cylinder head, and passes to a simple spraying nozzle. The power is controlled by varying the duration of the opening of the spray valve, and at the same time adjusting the pump to the pressure required. Years of trial have been necessary to obtain the results now given, but in

its present development the injection is reduced to the simplest possible terms. The result is a system which can be run with little previous knowledge and with a minimum of attention or danger due to carelessness in adjusting. The consumption per brake horsepower is about 0.4 pound per hour, figures as low as 0.381 at full power on official trials independently recorded being obtained, even in war-time hurry of delivery.

One or two points in the design may be noted. The first is the extreme accessibility gained by the plate column construction. This renders it possible for engines to be maintained and refitted by ship or depot staffs without calling in dockyard or manufacturers' assistance. The second is the conservative rating and stresses. Thereby extreme durability is obtained and serious breakdowns are avoided. The weight per brake horsepower of an engine should always be considered in connection with the rating and stresses allowed. The Vickers' construction is in principle a very light one, and even with the low rating results in a weight per brake horsepower of 58 pounds for the bare engine, calling the power 1,200 for the 12-cylinder engine. This weight is reduced to below 50 pounds if the power be taken as 1,400 horsepower, which is easily obtained, and would be very materially decreased if the piston speed were increased to the high figure of 1,323 feet per minute, as is obtained in some of the German "U" boat engines. The third notable feature is the avoidance of special methods of manufacture. While the best material and workmanship are used, special steels are avoided, as also are heat treatment of details and parts requiring very special tools or processes of manufacture. Thus spur wheels are used wherever possible in preference to spirals. The engine thus becomes an ordinary manufacturing and refitting proposition, and no danger arises if a broken part is replaced in ordinary material. Such a policy permits boats to be sent to foreign stations with more assurance than if they were dependent for upkeep upon expert supervision during repair.

Finally, are simplicity and safety. All adjustments are obvious and readily carried out, and the abolition of the air compressor renders the engine particularly simple to understand and to run.

The main propelling machinery of the "K" boats consists of two sets of single reduction geared steam turbines and two boilers of the straight-tube three-drum type, having a working pressure of 235 pounds per square inch and arranged for burning oil fuel. Each set of turbines consists of one high-pressure and one low-pressure cylinder, both turbines driving the main line of shafting by double helical gearing. An astern turbine is incorporated in each low-pressure turbine. The full power ahead shaft speed is 400 R.P.M., the corresponding turbine speeds being, high pressure 3,500 and low pressure 2,800. The main motors, which are in the wings of the ship, somewhat above the shaft centers, are also connected to the propeller shafting by double helical gearing, the ratio in this case being 2 to 1. The turbines can be declutched from the line of shafting when desired, while clutches are also fitted between the main motors and the shaft line. This latter set of clutches must be disengaged when the main shaft is running at above 220 R.P.M., and this unclutching is carried out by hand or automatically by governor gear. The turbines are installed in a separate compartment, water-tight bulkheads being provided at the forward and after end of the compartment, with two doors in the former, one for access to the boiler room and the other to the main passage way to the forward part of the vessel; and one door in the aft bulkhead leading to the motor room.

The boiler room contains, in addition to the boilers, the feed pumps, oil-fuel pumps, heaters and filters, and forced-draught fans, the latter being driven by impulse steam turbines. A hinged funnel is provided for each boiler, arranged so that it may be lowered into the superstructure and the

opening closed by a strong steel cover, both operations being performed simultaneously by means of an electric motor placed in and operated from the turbine room. In later boats a hydraulic semi-rotary engine is used. As a precaution against accident in the event of the funnel covers being damaged, an additional valve of special design is fitted on the hull of the vessel at the base of the funnel uptake. The covers for making water-tight the air vents in the boiler rooms are hydraulically operated from the boiler room. In later boats of the class these covers are in duplicate. Arrangements are provided for shutting off the supply of oil fuel to the boilers before the funnel opening can be closed. The turbines, boilers and all hot surfaces are carefully and thoroughly lagged with incombustible, non-conducting material, with the object of reducing the temperatures in the engine room and boiler room to a minimum, and an efficient system of ventilation is provided throughout the machinery compartments.

For cruising, it is possible to use an eight-cylinder 800 brake horsepower oil engine of the submarine type. This engine drives a dynamo which can be used for charging the batteries and for driving the ship at cruising speed by electrical transmission to the main motors. The auxiliaries for the oil engines are on the same lines as those fitted in ordinary submarine practice.

The auxiliary machinery in the ship is of the usual pattern for submarine service, though somewhat larger in cases. Two compressors are fitted for charging the 2,500 pounds high-pressure air bottles, of which over 100 are fitted in the vessel. One of these compressors is motor driven and the other is direct driven from the end of the generator shaft. Two low-pressure air compressors are fitted for supplying air for blowing the water from the main ballast tanks when the vessel has broken surface. Two three-throw double-acting reciprocating bilge pumps, driven by electric motors, are also fitted for pumping the tanks and bilges. The telegraphs are of the ordinary mechanical type fitted with electric bell replies.

A special feature of these boats is the fitting of hydraulic power for various operations, such as working vent valves in the ballast tanks, air intakes in the boiler room, and the raising and lowering of the periscope rams and wireless masts. In the later boats this system is also applied to the funnel covers. The steering gear, and also the forward and after hydroplane diving gears, are operated by means of motor-driven Variable-Speed Gear Company's hydro-electric units, one for each service, controlled from pedestals placed in the control room. Hand gear is fitted in each case for emergency use. The steering gear itself is of the usual submarine screw-gear type.

The forward hydroplanes are of the housing type, the planes sliding inboard along the shaft into the superstructure to protect them from heavy seas and during high-speed running on the surface. This operation is effected by hydraulic power, but hand emergency gear is also fitted.

The torpedo equipment of these vessels consists of four 18-inch torpedo tubes in the bow and four 18-inch broadside tubes. In later boats of the class the bow armament equipment consists of six 21-inch torpedo tubes.

This history of submarine boat and engine design and construction when carefully examined shows that the underlying principle has always been the carrying of weapons of offence. The torpedo is the weapon which has naturally developed as the submarine's particular equipment on account of the ease of firing when the boat is either submerged or awash. But guns of increasing caliber are being adopted.—"Engineering."

HEAVY OIL ENGINES FOR BRITISH SUBMARINE BOATS.

The 12-cylinder engine is one that has been made in large numbers during the war by Messrs. Vickers, both at their works at Barrow-in-Furness and at Ipswich. It has also been manufactured by a number of firms working under license from Messrs. Vickers. The latter firm has latterly introduced various improvements in detail and in arrangement of controls, and as these, owing to the necessity of rapid production, were not wholly incorporated in the engines made by licensed firms, who were working to Admiralty instructions, the description we now append will include points interesting even to those who are fully conversant with the earlier engines. This 12-cylinder engine is shown in elevation in Fig. 1, while a cross-section is given in Fig. 2.

The peculiar framing construction of the engines is plainly evidenced by these drawings. First of all, it will be noticed that each main bearing is carried by a separate steel casting, and these castings are recessed into longitudinal beams, athwartship fitted bolts, not shown in the drawings, tying the two together. This construction obviously entails considerable care in chocking the engines in the boat, but with trained workmen no difficulty arises, and the required accuracy is obtained in a surprisingly short time. In adaptations of these engines when weight was relatively unimportant, a cast bedplate has been substituted, and a similar design carried out in steel is, with modern foundry methods, applicable to the light weight engines, and would probably be proposed for future work where interchangeability with previous parts, as for Admiralty work, is not important. In this respect, as in many other points in the engine, some of which will be subsequently mentioned, the designers have been handicapped by their own success, as improvements which experience has pointed out to the firm were sometimes not acceptable on account of the necessity of maintaining uniformity with the large number of engines already in commission.

The columns consist of boiler plate somewhat less than thirteen-sixteenths inch in thickness. At the top is a forged steel head into which the column plate is rabetted, three rows of rivets securing the head to the column plate. The plate is cut with lightening holes, and at the bottom is branched by a slot cut for removal of the main bearing cup. To the lower extremities of these branches are riveted four forgings, forming the feet. These are of approximately triangular elevation, and are plainly visible in the sectional view. The riveting of the heads and feet of the column is most carefully carried out, the holes being reamered and the rivets inserted alternately from opposite sides to avoid bending the column. The column is stiffened by light angles on its outer edges, these angles forming the facing for light sheet steel plates forming the sides of the crank-case. Small circular hand doors are fitted in the outboard casings to facilitate feeling the liner and lower bearings. Further stiffness is given by angles plainly shown on the elevation which support the horizontal tie-plate below them. This tie-plate has a hole in it slightly larger than the cylinder jacket bottom flange, and supports the latter by means of four adjustable snugs bolted to the tie-plate. On the column head is placed a sandwich or distance-piece, which takes the weight of the liner, and to which the cover is secured by six 1½-inch studs. The connecting joint consists of a cast-iron ring ground into both liner and cover. The main impulse load is transmitted from the covers to the columns by two studs and four bolts, 1⅝-inch diameter, for each column between adjacent cylinders, two bolts only in addition to the studs being fitted to the end columns of each of the four groups of three cylinders, into which it will be noted the engine is divided.

This framing construction permits of close determination of stresses and lends itself to a light-weight engine, but the main practical advantage is in the extreme accessibility it gives to the crankhead and main bearings, while a cylinder is rapidly replaced should occasion arise. Accessibility and facility of overhaul are, of course, of the utmost importance for marine work.

The cover, it will be seen, is a simple steel casting, the inlet and exhaust valve boxes being separate from it. These boxes seat upon coned joints in the cover, a ground metal to metal joint preventing leakage from the compression space. They are very thoroughly water-cooled, as will be seen in the elevation, the inlet valve-box even having a recess turned on its exterior into which water from the cover has access.

The water joints between the top of the cover and the valve boxes are made by indiarubber rings compressed below the flange of the latter. While no trouble is experienced on service, yet in later engines Messrs. Vickers make this joint by a pair of rubber rings in grooves on the periphery of the box, which enter into a slight recess in the cover. The advantage of this is that the joint becomes a sliding one, automatically compensating for any wear due to grinding down of the valve-box. A similar water joint is fitted between the bottom of the liner and the jacket, the two grooves being visible in the illustrations.

The exhaust and induction valves are of nickel steel, the former being water-cooled by flexible indiarubber armored hose connected to the valve head. The inlet water passes down a central tube to the bottom of the exhaust valve and escapes upwards around this tube. The practice of water-cooled exhaust valves is followed by Messrs. Vickers even in their smallest submarine engines, one advantage being that the spindle clearances may be kept fine so that on starting the leakage of gas into the boat will be negligible compared with that from uncooled valve spindles in which a comparatively large clearance has to be allowed for expansion due to heat.

The separate valve-spindle guides will be seen in the section in Fig. 2, taken at the No. 11 cylinder, this being in accordance with the desire for extreme durability of all parts. Lubrication is not now fitted to the valve spindles. The head gear calls for no particular comment beyond stating that the spring spindle has a tee-end at its lower extremity, so that if it is depressed and then turned through a right angle about its axis it can be readily removed with its spring.

The cylinder proper is very elementary in form, and consists of a plain cast-iron liner surrounded by a corrugated sheet steel jacket. This latter terminates at its lower end in a pressing forging, to which it is welded, and at the top a flange is welded to the jacket and is jointed to the underside of the sandwich plate. The liner has a number of steel bosses riveted through it, as indicated in the elevation of the eight-cylinder engine (Fig. 3). These bosses are machined to form stuffing boxes through which the piston lubrication connections pass; these connections being screwed into the liner. Eight of these feeds are fitted to each cylinder, and are supplied by a sight-feed force lubricating pump. In practice this pump is set to a low output, as very little lubrication is required for the piston.

The piston is of simple design in cast-iron. The top is concave and fitted at the center with a screwed hole for a lifting bolt. This hole at one time held a special plug designed to maintain a high temperature, and thus to facilitate combustion, but this was found unnecessary and is not now fitted. Though the hole is left open to the flame, no harm is occasioned thereby, the main impact of the injection not coming upon the center of the piston as in an air-injection engine. There are six piston rings, 0.375 inch wide at the upper part of the piston, and one wiper ring at the bottom.

The gudgeon pin is pressed hollow from special case-hardened steel, a plug being subsequently fitted to exclude the oil. The diameter is in three steps, as is usual, and is tightly driven into the pisotn, being secured at one end by a set pin and at the other by a key. A light aluminum guard is fitted over the top of the connecting rod and is secured by the gudgeon-pin set screw and by a special stud at the opposite side. The somewhat complicated lower guard originally fitted has been abandoned.

The connecting rod is of plain turned design. The top end is spherical with a non-adjustable gudgeon bearing, consisting of a bronze bush pressed into the eye of the rod and lined with white metal. This bush has a radial hole through it at the middle of its length, this hole communicating with a groove turned in the eye of the rod. By this means, if the bush should by accident or carelessness turn slightly, the oil supply from the hollow rod is not interrupted. The lower end of the rod is forged into a palm in which the crankhead brass is spigoted, provision being made for a compression plate. The crankhead brasses are of bronze, white-metal lined, with a circumferential non-staggered oil groove at their middle length. The lubrication oil passes into the top of the main bearings, thence through the crankshaft to the crank-pin, from thence up the rod to the top end. It will be noticed that this design of crankhead is somewhat heavy, and Messrs. Vickers have adopted a modified design in cast steel for crankheads of their non-Admiralty work.

The crankshaft in the 12-cylinder engine is in four three-throw sections, and was maintained at the same diameter as the solid shaft in standard six, eight and ten-cylinder engines, namely 7½-inch. Nickel-chrome with hollow journals and pins was originally fitted throughout the large engine, mainly to save weight and to gain experience of this material in such shafts, the maximum stress not being materially affected by the addition of the extra cylinders. Difficulties in supply during the war led to this being, in a number of 12-cylinder engines, used only for the aft two lengths, the solid carbon steel pattern as originally used in the smaller engines being used for the forward lengths. All these shafts are oil-toughened, and the pins are trued by hand and lapped after turning.

The exhaust main is not shown on the drawing, but it consists of a pair of water-jacketed headers, one for each six cylinders, built up in lengths with independent outlets to atmosphere. The Admiralty engines are fitted with cast-iron mains, but for ordinary purposes welded sheet steel has proved quite satisfactory. The order of firing of cylinders is 1, 7, 3, 9, 2, 8, 6, 12, 4, 10, 5, 11, and deflecting diaphragms are fitted in the mains to prevent interference between the cylinder exhausts.

The flywheel is of cast-iron fitted with teeth on the outside to engage with a three horsepower electric or hand-turning gear of Vickers' patent, specially designed for instant operation, and to meet the limited space available. The center eye of the wheel is separate from the rim to which it is bolted by a series of bolts shown in Fig. 3, and it is divided along its diameter to facilitate machinery overhaul in the boat.

The 12-cylinder engine illustrated is of the latest class, in which the unit system of fuel pumps originally favored by the submarine service is fitted. In this the inlet and exhaust valve cams are fitted on a shaft running in ring-oiled bearings, at about the middle height of the engine. The lower end of each of the long push rods carries the case-hardened steel roller, and on either side of it is a guide block sliding upon specially prepared guides in the cam casing. A loosely fitted cover surrounds the push rod, and excludes dirt from the casing. A lead from the bearing oil supply is led to each cam roller and guide, the oil entering at the back and squirting into the hollow interior of the roller block, as shown in Fig. 2. The cam casings drain back to the crank-case, the only oil remaining in them being

that in the small troughs under each cam into which the latter dip. The cams are wide, and are made of case-hardened steel pressed to a hollow section. The cam-cases—which were originally of aluminum alloy, but owing to other demands for that material have during recent years been made in cast iron—are bolted to the camshaft brackets, and are provided with a footstep to facilitate access to the top gear. The lower camshaft is driven by spur gearing in the middle bay of the engine, while the upper camshaft receives its motion through a bevel-driven vertical shaft visible in the middle of Fig. 1. The timing gear on the vertical shaft is also shown on this illustration. On rotating the horizontal wheel it is moved vertically upon the thread shown, and thereby advances or retards the upper shaft relative to the crankshaft owing to the upper bevel wheel being driven through a number of spiral keys on a coupling moved vertically by the hand wheel and sliding on straight keys on the vertical shaft. As the upper shaft carries the fuel admission cams, this effect is analogous to that of moving the "spark" in a petrol engine. It is a refinement not usually applied to Diesel engines, but intelligently used it enables the best setting for any condition of working to be obtained.

For ordinary running the spray valve, when at full power, is due to be opened at about 16 degrees, measured on the crankshaft, before the top dead center of its piston, and the controls, which will be presently described, have been re-designed so that as the duration of the spray valve opening is reduced for lower powers; the point of injection is simultaneously adjusted to that suitable for ordinary surface conditions. With these conditions it is unnecessary to adjust the timing gear when varying the power, but in the event of a heavy "charge" being put upon the main motors of the boat, thus considerably increasing the load on the engine, it is possible to retard the injection a few degrees or with a light condition of the boat or a following wind the timing may be advanced slightly. Adjustments of fuel valve settings to suit variations of fuel are also very conveniently carried out in similar manner.

Messrs. Vickers have carried out extensive trials in various conditions to investigate the effect of the timing gear and the result is to show that within the limits of practical working the brake horsepower and maximum cylinder pressure are both increased by advancing the injection, the curve of increase on a base representing the angle of injection being a straight line. Indicator cards varying in character from the flat-topped card representative of the normal air injection engine to the sharply-peaked card of the explosion engine, can be obtained at will, and the whole forms a most interesting study. In ordinary trials the firm limits the maximum pressure to a more reasonable figure, but on occasions on service where extra speed has been required the engines have been run with timing gear adjusted to give cylinder pressures of 700 pounds per square inch and over without any ill-effect. The control of the spray valves is effected by the left-hand vertical hand wheel seen in Fig. 1. This, by gearing, partially rotates the spray valve control shaft, the lowest of the three shafts running along the top of the engine. At each cylinder this shaft has a notched lever keyed to it. Alongside this lever is a similar lever with a spring clutch handle, the second lever being fixed to a short sleeve riding on the spray valve control shaft. When this sleeve is rotated by the handle the eccentric fulcrum mounting of the spray valve lever is partially turned by means of the two levers and the connecting rod, plainly seen in Fig. 2. The eccentricity of this mounting and the link proportions are so designed that a close approximation to the required point of fuel admission to suit any reduced duration of admission is obtained. Ordinarily the clutch handles are pushed forward to engage with the levers on the control shaft, in which case all spray

valves are controlled by the hand wheel, and over a wide range of power a movement of this wheel is all that is required to regulate the power. Should it be desired to cut out any cylinder the handle is drawn down, disengaging the sleeve lever from the one alongside it, and on bringing the handle forward to a notch in the fixed quadrant, as shown dotted in Fig. 2, the spray valve is put out of operation, a small cam on the sleeve lifting the suction valve of the fuel pump and putting it also out of action as the spray valve ceases to operate.

The whole is particularly simple, though the repetition of the units for each cylinder gives a first impression of complication, but it is plain that experience was necessary to enable the required somewhat complicated co-relation of functions of timing and duration to be obtained by such an elementary form of apparatus.

The fuel pumps at each cylinder are eccentric operated and consist of bronze castings supported in a steel casting secured to the cylinder cover. This casting carries the upper camshaft bearing, on the cap of which is mounted a separate small bearing in which the top-most shaft slides. This shaft is actuated by the right-hand wheel in the center of the engine, and by means of a downwardly-projecting stirrup slides a spiral scroll cam at each pump to the position required for the correct fuel output. This output is varied as the cam does not allow the suction valve of the pump to close till the desired point in the downward stroke of the latter. The high-pressure pump is marked on No. 12 cylinder in Fig. 1, and the spring returning the tappet for the suction valve can be detected in the drawing. The bore of the pump is ½ inch and its stroke 1 inch, special soft packing being used in the gland. Ordinary webbed cone-seated valves are usually fitted, but ball valves have been found equally efficient, and are now finding favor in the service. The whole of the bracket, pump and details have been redesigned in Vickers' later models with advantages as regards ease of manufacture, weight and efficiency, but in so doing, interchangeability has been maintained to a considerable degree of detail.

The fuel from the pump passes through the pipe shown on the right of it to a strainer, and thence into an accumulator tube shown vertically just to the left of the right-hand column on No. 12 cylinder. It then passes to the spray valve. The strainer at one time consisted of a plate with fine holes drilled in it, but the later type, which is much cheaper and affords greater filtering area, consists of a series of perforated discs, each with a shallow circular projection on one face close to the edge. This projecting annulus is knurled, and a number of these discs strung on a central bolt form a cartridge with circumferential rows of tiny triangular holes on its surface, through which the fuel passes to the center and thence through the end plate to the outlet pipe. A vent valve is fitted to the spray valve body, this valve, for convenience, being fitted on the head of the pulsator, which casting also forms the strainer body. The pulsator is a slightly flattened steel tube, and its function is to receive the fuel charge from the pump, when this is supplying only its own cylinder, pending the opening of the spray valve. A pressure gage is mounted on the discharge pipe so that the pumping plant for each cylinder is complete in itself, as originally specified. At the bottom of the pulsator a valve is now fitted, and through this valve each pulsator can be connected to a main, originally fitted for priming purposes. The common practice is now to run with the pulsators coupled to this main, in which case the pumps are, of course, not measuring the fuel to each cylinder, the spray valves performing this function. This is opposed to the ordinary idea; but consideration will show that the valve is more likely to be accurate than the pump. The system thus run becomes a pressure

system, and all that is necessary is to maintain the required pressure and to adjust the spray valve functions to regulate the power. A development of this system will be seen when the eight-cylinder reversing engine in Fig. 3 is described.

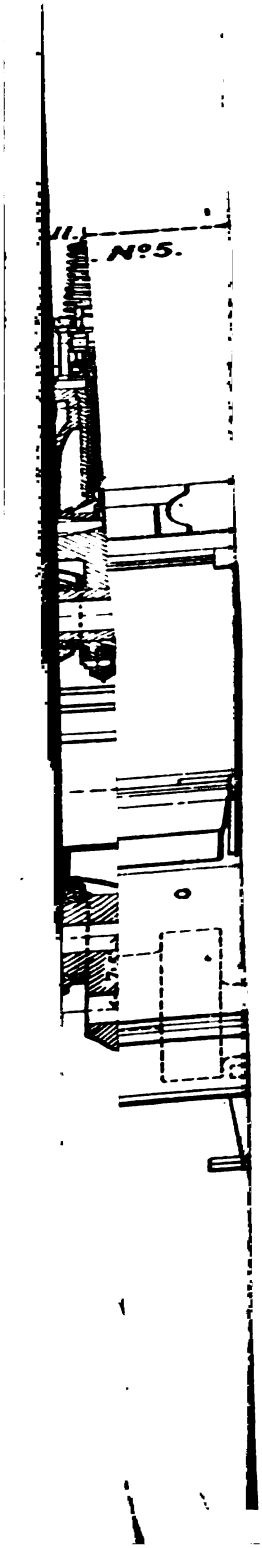
A safety valve is fitted to the pressure rail in the center of the engine, as is also a quick opening valve for immediately stopping the engine in case of emergency by releasing the fuel pressure.

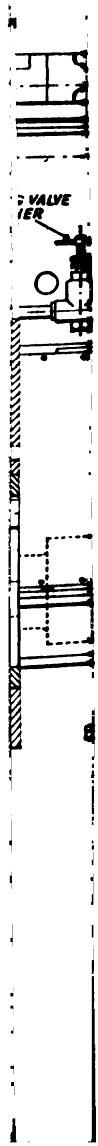
The spray valve in the standard engine is an ordinary spring-loaded conical valve, made in hard steel. The gland is packed with fuel-pump packing and, with reasonable care in first packing, is perfectly satisfactory. The valves can be individually adjusted by the nuts seen on the end of the spindle. The nozzle is of tool steel and its securing nut is bedded upon the coned seat in the cylinder cover so that there is no tendency for fuel to leak through joints to the cylinder. The number and disposition of the holes in the nozzle depend upon circumstances, but in the engine described they are five in number, and 0.0205 inch in diameter, the jets being at 68 degrees with the axis of the cylinder.

The lower part of the spray valve-box is a steel stamping, the upper part, which contains the spring and the guide plunger upon which it bears, is of cast steel, the eccentrically-bored fulcrum bearing of the spray-valve lever being mounted between the two parts. The spray valve is operated by a lever of the first order, shown in Fig. 2, which has a wide roller loosely mounted in a groove at its inboard end, this roller engaging with the fuel cam. The fuel cam consists of a small hardened toe-piece mounted in a steel cam disc on the upper camshaft and at the center line of the cylinder, and is capable of slight circumferential adjustment on its disc. A sheet save-all is fitted under each pump, and by means of a drain pipe avoids waste of fuel when cleaning filters or venting the system.

The engine is started by opening a T-handled air master valve, situated near the center of the engine, which admits air to the two starting air boxes between the end groups of cylinders. In these boxes are small partially-balanced valves, normally kept off their seats, and thereby free of the cams, by a light spring. When air is admitted the valves are blown down upon their seats and are operated in correct sequence by cams on a pair of athwartship shafts bevel driven from the lower camshafts. At one time air starting was regarded as an emergency measure, the electric motor being ordinarily used in submarines. Air starting valves fitted with hardened ends engaging with the cams were satisfactory in such circumstances. Nowadays air starting is recognized as safer in the event of water having entered the engine during driving, and the air-starting valves have been provided with hardened steel rollers to give increased durability, a very neat little design having been adopted. The air passes from the air-starting box to the individual cylinders, passing into the cylinders through a spring-loaded non-return valve on the cover. As seen in Fig. 2, it passes through small holes after leaving the non-return valve so that no broken part can enter the cylinder in case of breakage of a valve. The engine having started on air the spray valves are put in gear, and afterwards, when convenient, the air master valve is closed. With pipes and pumps fully primed, firing begins within the first two or three revolutions, the absence of the chilling effect of the injection air doubtless accounting for the ease in starting.

At the back of the engine is a bearing oil pump, of double-acting type, and a plunger pump for maintaining the gravity fuel tank full. These pumps run at half engine speed, and are driven by bevels from one of the starting air camshafts. In some boats where automatic compensation of fuel by sea water pressure is fitted, the gravity fuel pump is discon-


Nº5.


VALVE
IER

nected. A tendency is growing to accept separately-driven pumps in submarines, and if this matures the arrangement of any engine will be greatly simplified, the necessity of finding an accessible site on the engine for the auxiliaries presenting considerable difficulties where space is a primary consideration. The indicator shafts are also driven from the air camshafts. These consist of a shaft for each set of six cylinders, driven at engine speed, upon which are mounted adjustable eccentrics whose straps and rods move a vertical plunger at each cylinder to which the wire actuating the indicator is attached. A clutch with special tooth causes the indicator shafts to revolve when cards are to be taken. By this method the crank pits are kept clear of all parts likely to fail on long runs. A pair of small lubricators of plunger type are also fitted at the back of the engine. These supply oil to the pistons as previously described.

The induction header is not shown on the drawings, but it consists of a light steel pipe for each half of the engine, to which the induction valve boxes are coupled by trumpet-shaped pipes. The air to each header passes from the atmosphere through a bell-mouthed inlet, while a lead is also arranged to take a small air supply from the crank-case. This latter supply leaves the crank-case through a hole in each tie-plate and passes through a light fiber non-return valve on its way to the induction header. By this means the crank-case, which has small inlet grids at the ends of the engine and in the center, is maintained under a slight vacuum, and the atmosphere in the engine-room is not contaminated by the egress of vapor when running or after stopping.

The circulating water is led from an independent rotary pump to a main supplying the jackets, from which it passes to the cover through six tubes through the sandwich plate, the joint round these tubes being made by vulcanized rubber rings. It then passes to the jacket of the exhaust bend and away. The exhaust valves are fed from a separate main with a filter at either end. A short-circuiting valve is fitted between the pump suction and the cooling water discharge, so that in the event of a sudden dive to a great depth being made, the sea valves may be closed, and yet a circulation of the water in the closed system can be maintained till the engines have cooled.

The fuel oil flows from the gravity tank, which is placed in any convenient place in the engine-room, to the center of the engine, passing first through a settling pot where sediment and water can be removed, and then to the middle of a main running the length of the engine. This connection is so arranged that surging from end to end in a seaway is prevented, and from it the main rises towards each end facilitating venting and enabling a standard main to be used whether the rake of the engine in the boat is up or down. From this main small pipes with shut-off cocks lead to each high-pressure fuel pump, the fuel being filtered by a fine gauze filter embodied in the suction chamber of the pump. The pump itself, with its discharge filter, has been described above.

Examination of the drawings will show that all of the parts of the engine are particularly accessible, while in the construction ease of overhaul and adjustment has been carefully studied. To this and to the moderate rating of 1,200 brake horsepower to 1,400 brake horsepower, at 380 R.P.M. to 400 R.P.M., may be attributed the long and successful running that so many of the British boats have carried out without the necessity of assistance.

We are able to give, in Fig. 5, a copy of a typical indicator diagram from one of these engines, together with the record of a 6-hours' full

power trial (Table No. I) independently recorded, which will doubtless be of interest in view of the many discussions regarding the efficiency of solid injection.

TABLE I.—SIX HOURS' OFFICIAL SHOP TRIAL OF SUBMARINE ENGINE 1,200 B.H.P. DATED SEPTEMBER 5, 1918.

Time.	R.P.M.	B.H.P.	Fuel Used.		Fuel Pressure.	Controls at		Circulating Water.	Lubricating Oil.			Atmosphere Temperature.
			Lb. per Hour.	Lb. per B.H.P.-Hr.		Fuel Pump.	Spray Valves.	Pressure.	Pressure.	Temperature, Fahrenheit. In.	Temperature, Fahrenheit. Out.	
					lb. per sq. in.			lb. per sq. in.	lb. per sq. in.	deg.	deg.	deg.
1st hour	379·3	1,209·9	461	0·381	4,500	9	18·5	10	12	68	87	60
2nd hour	382·5	1,218·5	461	0·379	4,500	9	18·5	10	12	71	101	62
3rd hour	381·0	1,213·9	461	0·380	4,500	9	18·5	8	10	72	106	64
4th hour	381·8	1,216·3	457	0·376	4,500	9	18·5	8	8	87	109	62
5th hour	381·4	1,215·0	479	0·394	4,500	9	18·5	10	12	98	116	63
6th hour	381·7	1,216·0	461	0·380	4,500	9	18·5	10	12	104	114	63
Average	381·3	1,215·0	463	0·381	—	—	—	—	—	—	—	—

Indicated horse-power, 1520.
Inspecting Officer's Report: "The engine ran steadily and well, and no trouble of any sort was experienced throughout the trial. The crank pits remained clear of gases, and the exhaust was light and good. The trial is considered satisfactory."

Fig. 3 shows an eight-cylinder reversing engine in elevation, Fig. 4 giving an end view. A number of these engines were ordered by the Russian Government from Messrs. Vickers for patrol vessels. The cylinder dimensions are the same as in the twelve-cylinder engine, namely,

$14\frac{1}{2}$ inches diameter by 15 inches stroke, the engine easily developing 800 brake horsepower at 380 R.P.M. It will be noticed that the construction of the larger engine has been closely followed whenever possible, but in this instance the engine has been designed to operate on the common fuel pump or "rail" system. The fuel pumps are shown in the end view and are driven by spiral wheels from the end of the crankshaft. There are four plungers, 0.66 inch diameter by 1.25 inches stroke, the revolutions being 190 at full engine speed. Each pair of pumps can be isolated from the other or put out of action if required. A separate discharge is led to the control box in the center of the engine, so that if required one pair of pumps may serve one group of four cylinders. By these means, total stoppage in the event of failure of either pump or fuel main is prevented. As a matter of fact, these pumps, with their double-suction valves and forced lubrication are wonderfully reliable, while failure of fuel piping is almost unknown, the high pressure of sometimes over 4,000 pounds per square inch presenting no more difficulty than lower pressure in thinner piping.

The smaller hand wheel is free to be rotated when the reversing gear is in the full ahead or astern position. Its first turn opens a master valve in the control box which passes air to the air distributor boxes at the ends of the engine. These are similar in principle to those on the twelve-cylinder engine, but are disposed radially in a neat bronze casting, one cam only in the box being required for each direction of running. Further turns admit fuel to the main supplying first one group of four cylinders and then the remainder, the air being finally shut off. A last turn stops the engine by shutting fuel off the mains. The fuel passes through a strainer and a shut-off valve to the spray valve. Quick-closing shut-off valves are fitted in later engines of this type, the handle by its position serving as an indicator as to whether the supply is open to the spray valve. A small lever which may be seen at the running position is connected to the fuel pump, and by modifying the operation of the mechanically-operated suction valves in the usual manner, regulates the pressure of the fuel supply, and thence at any setting of the spray valve determines the amount injected.

The spray valve regulation is effected by the lever at the top of the engine in the middle. This lever rotates a shaft passing through bearings on each cover, the intermediate fuel levers being mounted on eccentrics, on this shaft. This intermediate lever is moved through the intermediary of a short tappet rod by the fuel cam on the single camshaft, which in this instance is mounted on brackets attached to the top of the columns and carries the inlet and exhaust cams also. The other end of this intermediate fuel lever depresses the bell crank of the spray valve, a roller in the lever sliding upon a hardened piece in the bell crank.* The spray valve in these engines differs from that of the twelve-cylinder engine, the spindle being totally enclosed within a steel stamping and operated by a bell crank, the spindle of which passes through a gland in the side of the valve casing, a ball race taking the unbalanced fuel pressure on the bell crank. This design was chosen as lending itself better to the upper camshaft arrangement than did the standard valve.

The reversing of the engine is carried out by the large hand wheel, which, through a scroll and crank, first by partially turning a reversing shaft—mounted in the same bracket as the camshaft, but above it—swings the inlet and exhaust push rods clear of their cams and then, after sliding the camshaft till the reverse cam is in the correct position, replaces the push rods in their working position. This hand wheel is back-locked with the starting wheel so that it cannot be operated except when the latter is in the "stop" position. The fuel and air-starting cams being of small

lift can be pushed past their respective rollers without the latter being otherwise lifted, the cams and rollers being tapered to facilitate this operation.

TABLE II.—*Engine No. 521 Port. Official Starting and Reversing Trial, April 26, 1919.*

From.	To.	Condition of Engine.	Volume of Compressed Air in Use.	Air Pressure. Before.	Air Pressure. After.	Air Pressure. Difference.	Time to Start.	Revolutions to Start.
			c. ft.	lb.	lb.	lb.	sec.	
Stop	Ahead	Cold	36·4	550	525	25	5	8·0
Stop	Ahead	Cold	36·4	525	510	15	3	4·0
Stop	Ahead	Cold	26·4	510	495	15	3	4·0
Stop	Ahead	Cold	36·4	495	485	10	3	4·0
Stop	Astern	Cold	36·4	480	465	15	2	3·0
Stop	Astern	Cold	36·4	465	450	15	2	2·5
Stop	Astern	Cold	36·4	450	440	10	2	2·5
Stop	Astern	Cold	36·4	440	425	15	2	2·5
Astern	Ahead	Slightly Warm.	36·4	400	380	20	15	—
Ahead	Astern		36·4	380	365	15	16	—
Astern	Ahead		36·4	365	355	10	16	—
Ahead	Astern		36·4	355	345	10	15	—

Note.—Load condition as at full power.

The fuel system was primed by hand for the first start to approximately 400 lb. pressure. This was not necessary for the subsequent starts, as there was sufficient residual pressure in the system. For reversing no hand priming was carried out.

Of the time taken to reverse the engine, from 13 seconds to 14 seconds were absorbed in hand manipulation of shutting off and opening the controls and actuating the reversing wheel. Number of turns of reversing wheel from full ahead to full astern equals 23.

TABLE III.—*Engine No. 518 Port. July 3, 1918. Air Starting Trial Arranged for Lloyds Surveyor.*

Thirty-four starts were made, dropping the pressure from 450 lb. to 195 lb. per square inch. Volume of compressed air in use 37·5 cub. ft.

The engine was cold, not having been touched since the previous morning's preliminary air starting trial.

Load condition of engine as at full power.

Note.—The number of starts could have been increased, but the air bottles were some considerable distance from the engine, and due to the long length of pipe, considerable "wire drawing" took place.

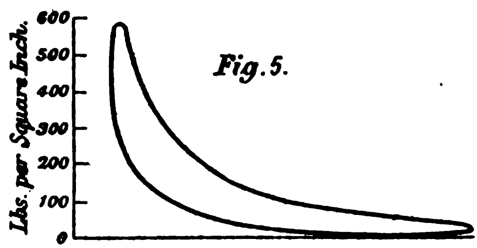


Fig. 5.

The drive for the camshaft in this engine is entirely by spur wheels, the sliding motion of the camshaft wheel being met by the provision of a wide wheel in the train. The bearing oil and gravity fuel pumps, together

with the usual indicator gear and piston oil pumps, are at the back of the engine in the center-bay. The two former pumps can be disconnected from the engine and worked by hand for priming purposes, a similar arrangement being followed in the twelve-cylinder engine.

Hand reversing was adopted instead of power gear on the ground of reliability after long service. Tables II and III, annexed, give the results of the starting and reversing tests, and incidentally show the rapidity of starting and the low consumption of air.

These engines gave great satisfaction to the Russians, but owing to the collapse of that country they did not receive them. A number were allocated to the Anglo-Saxon Petroleum Company, who used them for installation in some converted sailing ships. For this purpose, to suit the low speed of the ship, the propellers were designed to absorb 630 brake horsepower at 320 R.P.M., the engines being adjusted to suit. A number of these installations have been carried out, some in China, with satisfactory results.

These engines are interesting as showing a complete remodelling of type while maintaining interchangeability of important parts to a considerable extent. Attention to details in the re-design enable the weight of the reversing design to be brought below that of the eight-cylinder non-reversing design on which it was based, a few inches increase in overall length over the latter design having had to be taken to accommodate the reversing gear and central drive in the same bay.—"Engineering."

THE USE OF PULVERIZED COAL.

Under the auspices of the Fuel Research Board of the Department of Scientific and Industrial Research, Mr. Leonard C. Harvey, in the middle of last year, made a journey to the United States, in order to study the applications of pulverized coal to various uses in that country. On his return he made a report to the Board, which came to the conclusion that the information contained in that document would be of substantial value to the consumers of coal in British industries. It therefore published it as "Special Report No. 1," which is purchasable from agents for Government publications for the sum of 2s. 6d. net. Subsequently Mr. Harvey read before the annual meeting in May last of the Iron and Steel Institute a paper entitled "Use of Pulverized Coal, with Special Reference to its Application in Metallurgy," which was intended to form a supplement to the information contained in the report. Feeling sure that the matter will prove of interest to our readers, we propose to give in the following articles a summary of and extracts from these two documents. We may say before proceeding to do so, that both the report and the paper contain bibliographies of the subject, that in the former being the more comprehensive of the two.

At the outset it may be explained that Mr. Harvey visited twenty-three installations in which powdered coal was used commercially for, amongst other things, billet heating, puddling, ingot heating, tin-plating, annealing, steel, rivet heating, and reverberating melting furnaces, as well as for steam boilers, cement kilns, &c. He also investigated the following systems: The Quigley, Fuller, Holbeck, Muhlfeld, Pruden, Covert, Bergman, Aero, Kinyon and Stroud.

The report is divided into seven chapters, and has four appendices.

In Chapter I. Mr. Harvey, after quoting an article which appeared in the Engineering Supplement of "The Times," in February of last year, and which discussed the principles underlying the use of fuel in a pulverized form, first of all reviews in outline the several operations necessary

in order to obtain a sufficient degree of pulverization of the fuel. He then considers the range of fuels suitable for use in pulverized form, referring in succession to coke, pitch, anthracite, bituminous coals, lignite and peat.

In Chapter II. there is, first of all, a table which shows the original cost of pulverizing plants of varying sizes. It was compiled from figures furnished by the Fuller Engineering Company, of Church Street, New York, and gives in parallel columns the pre-war and estimated present-day costs of plants having outputs of from 10 to 250 tons per day. For our present purpose we need only concern ourselves with the prices now ruling. It appears that a plant, with a capacity of from 10 to 40 tons per day, is estimated to cost £9687, and will necessitate the use of one 33-inch mill. A plant capable of turning out from 50 to 90 tons per day will cost £11,562 and will require one 42-inch mill. A 100 to 130-ton plant will cost £14,062, and will need three 33-inch mills; a 140 to 180-ton plant, having two 42-inch mills, will cost £15,625; while a 190 to 250-ton plant with three 42-inch mills, will necessitate an outlay of £19,375.

As regards the cost of pulverizing the coal Mr. Harvey quotes the experience of three separate works. In the first case the possible output of the works was 80 tons and about 30 tons were actually produced. In the second case the capacity was the same—80 tons—and the output from 30 to 35 tons, while in the third the full capacity was 50 tons, and about 30 tons were produced. In case No. 1 the outgoings taken into account were for oil and grease, waste, stores and fuel for drying, labor, current at "0.02 cent"—evidently a mistake for dollars—or about 1d. per kilowatt-hour, machine shop charges, interest and taxes. The total cost, taken over a period of one month, worked out at 2.21 dols., or, say, 9s. 2½d. per ton. In case No. 2 the operating costs given include labor and labor insurance, stores, electric repairs, machinery repairs, engineering repairs, and electric power at 0.018 dols. No allowance was therefore made for interest and depreciation. The cost averaged over a period of six months 1.48 dols., say, 6s. 2d. per ton. If, however, interest and depreciation at per cent on £11,562—the average cost of a plant to produce from 50 to 90 tons per day—be added, the total cost works out at 2.34 dols., or 9s. 9d. per ton. In case No. 3 the items taken included direct labor; repairs, labor; repairs, material; supplies, waste, &c.; insurance; and on-charges, and the total arrived at, as the average taken over a month's working, was 2.25 dols., or, say, 9s. 4½d. per ton. As a matter of fact, this total was higher than it would ordinarily be because during the month there was an unusual expenditure on repairs, &c., and the normal cost is given as being more nearly 1.489 dols., or, say, 6s. 2½d. per ton. Taking this lower cost the average for the first three plants works out at 2.013 dols., or just over 8s. 4½d. per ton of pulverized coal delivered to the furnaces.

The foregoing figures relate to present-day costs. For purposes of comparison Mr. Harvey gives the cost during the year 1913, and therefore before prices were influenced by the war, of a considerably larger plant, which had a full capacity of 150 tons a day, and the output of which was 140 tons per day. The costs for fuel for drying, operating, power—steam and electric—repairs to machinery, and repairs to buildings, came out at 0.602 dols., or just over 2s. 6d. per ton. Dividing 14 per cent on the capital outlay of £10,417 by the total yearly production—42,000 tons—and adding the result to the 2s. 6d., the figure of just over 3s. 2½d. per ton is arrived at. This figure is not strictly comparable with those of the first three plants, since, whereas two of them were working very considerably below their full capacity, and the third at only 60 per cent of its full capacity, plant No. 4 was producing over 93 per cent of the quantity for which it was designed. However, the items of cost, which in all

cases are given in detail, will doubtless prove to be useful guides as to past and present costs in the United States under the specific conditions mentioned.

The report then goes on to outline the various processes through which the coal passes in standard mill-house practice in American installations before it becomes pulverized fuel. The run of mine, lump coal or screenings, are delivered alongside the pulverizing mill in railway trucks, fitted preferably with bottom dumping doors, so that the coal may be discharged into track hoppers. From the latter the coal passes first of all to a crusher, in order that it may be broken down so as to pass through a screen of ¾-inch mesh, and is taken thence to the crushed coal bin, past a magnetic separator. From the crushed coal bin the coal descends by gravity to the dryers, which are thus referred to by Mr. Harvey: "Dryers of several efficient types are in use. They are heated by means of hand-fired grates, mechanical stokers, or pulverized coal burners. The hot gases from the furnaces are sometimes passed directly over and in contact with the coal, or through a central tube to the end of the coal-drying cylinder, and thereafter returned along the outer space in contact with the coal, or again, a compromise is made between these two systems. For the latter case, the hot gases first come in contact with the outer shell of the dryer to about half-way along its length; they are then collected and transferred to the end of the cylinder where they enter into direct contact with the coal as the gases pass back through the cylinder to the furnace end, where they escape to the dryer chimney. As a rule the dryers are rated to comply closely with the maximum capacity of mills operating. Crushers are generally started up in advance of the pulverizing mill, and are run for a short time after the mills are closed down. In this way a sufficient amount of dry coal is available when the mills are started up. Coal should be dried to within one per cent moisture, about 10 per cent moisture being extracted at each passage of the coal through the dryer. When fuel contains a heavy percentage of water the coal can be passed through the dryer a second time." From the dryers the coal is taken to the top of the building by an elevator and delivered into the dry coal bins.

In addition to the several types of grinding mills used in cement works, the mills generally installed in American steel works are the "Fuller," the "Raymond," and the "Bonnot." The two last mentioned are ordinarily run with air separator fans. The Fuller mills have only been equipped with them comparatively recently. When they are not fixed the pulverized coal as it comes from the discharge spouts is taken up to the top of the mill-house by means of bucket elevators. The advantage of air-separation exhaust from the mills lies in the direct delivery of the coal dust to the bins without having to employ elevators. Moreover, there are the additional advantages of simplicity of plant, absence of dust leakage through crevices in the mills, and absence of dust from the elevator casing, because the elevator is non-existent, while, as a consequence, there are no repairs of the elevator to pay for. On the other hand, the total power for grinding and delivering the coal by air-separation is somewhat greater than that required to run screen mills and bucket elevators. Further than that, the coal dust must pass through the suction fan at high speed, with the result that the blades are rapidly worn, and it is necessary, moreover, to install cyclone separators to deposit the coal dust from the air current. Summing up, Mr. Harvey remarks: "With the exception of power absorbed, the advantages of one method seem almost to balance those of the other."

The standard of fineness that has been adopted throughout the United States for industrial coal dust plants is 85 per cent through a 200 mesh

screen—40,000 holes to the square inch—and 95 per cent through a 100 mesh screen—10,000 holes to the square inch. The coal as it leaves the dryer before grinding should not contain more than 1 per cent of moisture.

In Chapter IV. Mr. Harvey discusses the various methods of transporting the coal dust to the burner under the following headings:

(*a*) By screw conveyors.
(*b*) By the air pressure system.
(*c*) By the air mixture system.
(*d*) By the enclosed pipe cable system.

With regard to (*b*) it is explained that it was patented in 1916 by Messrs. Magarvey, Salton and Heisler, and that the process, which has been of late quite extensively adopted, is controlled by the Quigley Furnace Specialties Company, of 26, Courtland Street, New York City. In it what are called "Blowing Tanks" are installed in the mill-house. Compressed air is applied to these tanks for the purpose of forcing any given quantity of coal through small supply pipes to bins at the furnaces. The tanks are arranged on weighing platforms, so that the weight of coal delivered into them can be read off on scale dials. The coal dust travels through ordinary 3-inch or 4-inch screwed piping to the bins at the furnaces. In view of stoppage caused by moisture in the coal, or due to condensation within the conveying pipes, a small companion pipe with tappings into the coal delivery pipe is provided. From it compressed air can at any time be supplied through the tappings or "bleeders," as the makers have termed them, and any stoppage of coal broken up and cleared. Mr. Harvey remarks, concerning the system, that the advantages pertaining to it are considerable. By it the coal dust can be transported through relatively small pipes and normal bends, which can be led overhead or underground with just the same convenience as can water or gas pipes. An ingenious accessory to the system, designed for the semi-automatic feeding of coal dust to any number of bins throughout the works, is also discussed.

The "air mixture" method is that employed in the Holbeck patented system, which is controlled by the Bonnot Company of Canton, Ohio. In it the coal dust is mixed at the mill-house with approximately half the quantity of air required for its complete combustion. Supply is effected at a pressure of 10/12 ounces per square inch, and a speed of 90 feet per second through the main supply pipes to the branch service at the furnaces is reached. The main supply pipe makes a complete loop from and to the mill-house, at which point the returning coal is extracted in a cyclone separator and re-delivered to the pulverized storage bin. In cases in which long lengths of main supply piping have to be used—say, in a loop of 3,000 feet—probably one or two booster fans would be employed.

The "enclosed pipe cable system" consists of a complete circuit of delivery pipe inside which is run a central cable fitted with discs for the conveyance of the coal dust along the supply circuit. Side outlet discharge openings controlled by shut-off valves are provided at the points where the fuel is to be used. The system, Mr. Harvey explains, is only in the experimental stage, and will probably be confined to relatively short delivery distances.

On the question of spontaneous combustion Mr. Harvey observes that it is sound practice not to store more than forty-eight hours' supply of dried coal. Spontaneous combustion is not likely to occur in that time providing that the coal as delivered from the drier is not overheated. The temperature at which it leaves that apparatus should ordinarily not exceed 200 degrees Fah., but the permissible temperature depends upon the nature of the coal and its ash. Not more than a twenty-four hours'

supply of pulverized coal should be stored on the mill-house, and it is considered advisable to store only sufficient coal in the furnace bin to last for one shift—eight hours or so. It would not appear, however, that much difficulty has been experienced in America from spontaneous combustion in the mill-house. Mr. Harvey mentions cases in which as large a bulk as from 30 to 40 tons of pulverized coal had been stored for three or four weeks without any sign of spontaneous combustion. In the neighborhood of the furnace, too, bins holding 10 to 15 tons of pulverized coal have been known to remain charged for three or four weeks without ignition. On the other hand, fires have occasionally occurred overnight. The only effect of spontaneous combustion is, apparently, to produce a slight caking in the center of the bulk. The actual fire is ultimately self-extinguished in the absence of sufficient oxygen to support combustion and by the generation of carbon dioxide gas.

As regards explosions, Mr. Harvey states that at none of the works that he visited was it found that any special precautions had been taken against explosion of the coal dust other than those dictated by common sense. "Installations," he remarks, "had been in use without any explosion having taken place for so many years that the danger of explosion had almost been forgotten." In some of the older plants mills, bins, girders, stairs, elevators, &c., were thickly coated with coal dust, and, remarks Mr. Harvey, "an unnecessary risk of fire, if not of actual explosion, is being courted in several plants." However, he continues, "the existence of such conditions in the mill-house for years without mishap should reassure those who are excessively fearful of coal-dust explosions." At new plants apparently greater attention is being given to keeping the mills and mill-house clean, and Mr. Harvey adds, "In fact, the new types of mills and the care now given to design of plant makes for a dust-tight equipment." In storing powdered coal the following precautions are recommended: (1) Limit the height of the coal pile to 10 feet or 12 feet; and (2) isolate the coal from all sources of heat such as steam pipes, flues, reflecting surfaces and direct sunlight.

In Chapter V. illustrations—which we reproduce—are given of the Quigley, Lopulco, Bergman and Fuller and Pruden burners. It is explained that the present tendency is to use low-pressure air supply for mixing with and introducing the coal dust into the combustion chamber or furnace, thus allowing the particles of ash to settle out before reaching the work to be heated or the boiler tubes and surfaces. Burners are therefore, says Mr. Harvey, not as a rule of a complicated nature, and more often than not the coal dust is simply allowed to fall by gravity from the feed screw into a single pipe, which carries the air for the combustion of the fuel. The pressures used vary from 2 ounces up to 6 ounces per square inch, but the lower the pressure the better. One form of Quigley burner—see Fig. 1—has a double air supply, provided from one source. The primary air has an ejector action upon the supply of powdered coal. The initial mixture of coal dust and air is then introduced into the main air supply just prior to its entry into the furnace. The pressure of the air supply at the point of entry of the fuel is about 2 ounces to 3 ounces.

The Fuller burner—see Fig. 3—usually consists of a single air pipe or burner tube into which the coal dust falls by gravity from the screw feeder. The pressure of the air supply is generally 2 ounces per square inch.

The Bergman burner—see Fig. 5—is fitted with an internal cone arranged concentrically within the air supply pipe, and the coal dust is fed in by gravity in such a manner that it forms a complete film or

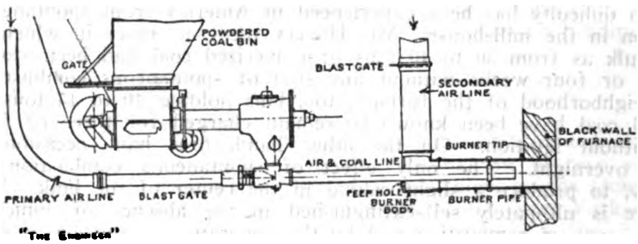


FIG. 1—ONE FORM OF QUIGLEY BURNER

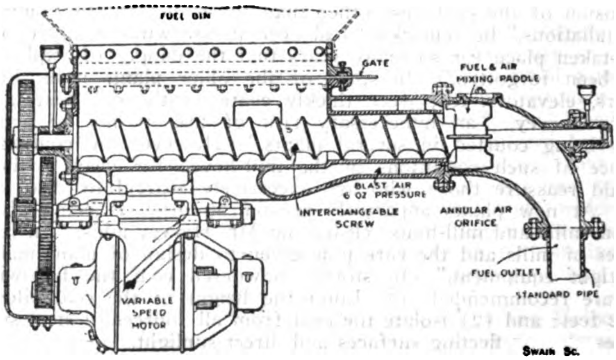


FIG. 2—THE LOPULCO BURNER

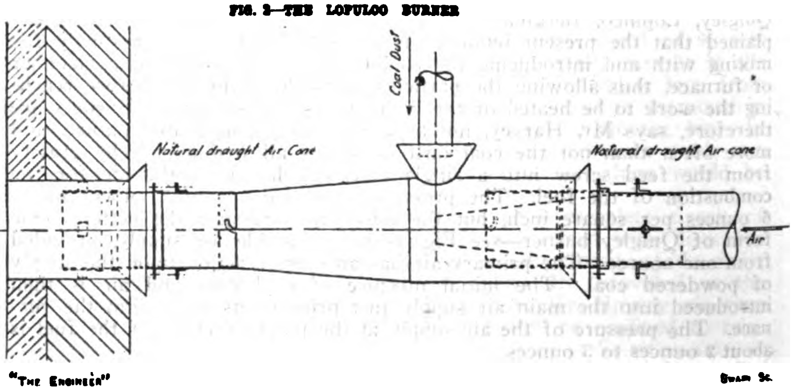
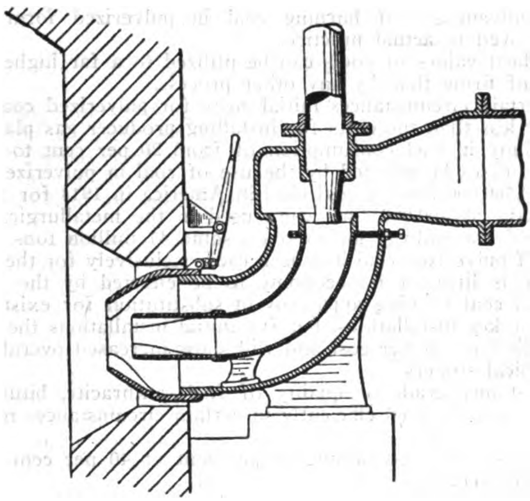


FIG. 3—THE FULLER BURNER

coating to the air jet, the purpose being to introduce the air into the furnace in the center of a covering of fuel. It is contended, says Mr. Harvey, that when coal dust is fed into the center of the air jet the latter expands away from the pulverized fuel and an imperfect mixing of the air and fuel in the furnace results.

The "Lopulco"—see Fig. 2—or Locomotive Pulverized Fuel Company's mixer, is combined with the screw feeder. The burner consists only of the pipe through which the mixed coal and air is conducted to the furnace. In practice, air is supplied at 6 ounces to 8 ounces pressure and enters the jacket surrounding the delivery end of the screw feed. At that point the mixed coal and air is given a whirling motion by means of the paddle blades, which assists the intimate mixing of the fuel and air for combustion. From the combination feeder-mixer a number of burner tubes can be supplied.



"THE ENGINEER" SWAIN SC.

FIG. 5—THE BERGMAN BURNER

With regard to other burners, Mr. Harvey states that the Heyl and Patterson Company, of Pittsburgh, which controls the "Covert" system, has several types of burners or mixers, one of them being of special interest. Into it the coal dust falls in the usual manner by gravity and descends on to diaphragm shelves projecting one below the other and fitted horizontally across the air inlet pipe or burner tube. Air is supplied at 2-ounce or 3-ounce pressure and picks up the coal dust on passing between the shelves. All the feeders hitherto referred to, with the exception of those of the Quigley type, operate with variable-speed motors for the control of the supply of coal dust to the burners. The Quigley feeder is fitted with adjustable jaws, through which the coal dust is passed to the burner. The feed screw can therefore rotate at a constant speed.

Finally, reference is made to the "Pruden" carburizer or mixer supplied by the Powdered Coal Equipment and Engineering Company, of Chicago. The apparatus effects, according to Mr. Harvey, perhaps the most perfect mixture of coal dust with air. The powdered coal, as it is withdrawn by a screw feeder from the bunker, is mixed with a primary supply of air and the mixture is then conveyed to a second chamber, in which the balance of air required for combustion is introduced into the primary mixture.

Chapter VI., which contains the opinions of numerous users of powdered coal fuel, applications of the system, working results, &c., is already in so condensed a form that it is practically impossible to compress it further. It is full of valuable and instructive matter, however, and those interested in the question of powdered coal fuel will be well repaid by obtaining a copy of the report and reading it *in extenso*.

Chapter VII., which deals with Mr. Harvey's conclusions, we reproduce in its entirety as follows:

(1) The advantages of burning coal in pulverized form have been definitely proved in actual practice.

(2) The heat values of coals can be utilized to a far higher degree by this means of firing than by any other process.

(3) In certain circumstances initial costs for pulverized coal plants are considerably less than the costs for installing producer gas plants.

(4) Economy in fuel consumption of from 20 per cent to 50 per cent can in many cases be effected by the use of coal in pulverized form.

Of the 513,500,000 tons of coal used in America in 1914 for all purposes, of which, say, 205,400,000 tons were used in the metallurgical and steel industries and for railway locomotives, some 41 million tons would have been saved if pulverized coal had been used exclusively for these purposes.

(5) There is little or no economy to be effected by the introduction of pulverized coal burning apparatus in substitution for existing efficient mechanical stoker installations, but for initial installations the latter plant can be installed at a lower cost and will show increased overall economies over mechanical stokers.

(6) Almost any grade or quality of fuel—anthracite, bituminous, lignite, or peat—can be used efficiently in certain circumstances in pulverized form.

(7) High ash fuels, containing 30 per cent or 40 per cent ash, can in some cases be used.

(8) Large quantities of what is considered waste coal can be used to good purpose, and culm and slack heaps at mines can be at once turned to profitable account.

(9) In suitably designed plants there is practically no danger whatever of the possible explosion of coal dust.

(10) With the precautions as to limited storage recommended, spontaneous combustion introduces no apparent difficulties.

(11) That the slag or dust resulting from the ash in coals can be conveniently and effectively handled and removed from all classes of melting and heat treatment furnaces, stationary boilers and locomotives, small furnaces such as rivet-heating furnaces being an exception.

(12) A very important start has been made in connection with the firing of large powerhouse boiler plants by this means, and its extension in this direction is likely to develop rapidly.

(13) In view of the attention now being given to marine boiler firing by this means, useful and important results are to be expected.

(14) Owing to the very considerably reduced amount of labor incidental to a pulverized coal plant as compared with hand firing and, in certain

cases, stoker firing, the labor saving is a most important feature introduced by this system of burning coal.

Saving in labor is particularly marked in connection with the firing of railway locomotives by this means.

(15) In view of the smokeless combustion of pulverized coal in metallurgical furnaces, and especially in the steel industries, for boiler firing and for locomotives, the abatement of smoke nuisances in large cities by this means can be accomplished to an appreciable extent.

Appendix I. gives a list of the systems investigated in the United States by Mr. Harvey, with the addresses of their proprietors. Appendix II. is devoted to a list of installations visited, with brief details in each case of the particular direction in which powdered fuel is applied. Appendix III. contains a fairly extensive list of powdered fuel users, while Appendix IV. embodies the Bibliography.—"The Engineer."

APPLICATION OF ELECTRICAL ENERGY TO THE MELTING OF METALS.*

By H. A. Greaves.

The conversion of electrical energy into heat presents a number of problems differing completely from those connected with the production of heat by the usual combustion processes, and owing to the almost unlimited temperatures obtainable electrically, and the absence of combustion products, a new field of development has been opened up to engineers and metallurgists.

The generation of heat electrically is almost entirely carried out by passing a current of electricity through suitable material. As everyone knows, when a current of electricity is transmitted by any substance, a certain amount of heat is generated in the substance depending mainly on the electrical resistance of the material and the current flowing.

Owing to the comparatively low electrical resistance of most metals, it is not feasible, when melting metal in bulk, to increase the current to such a value that the metal reaches a melting temperature by its resistance loss only, except in furnaces of the induction type. The induction furnace melts the metal by resistance of the charge, and the enormous current necessary are obtained by arranging the charge in the form of a ring round the iron core of a transformer. This type of furnace is not commercially satisfactory, owing to—

1. The loss in efficiency caused by the metal being in the form of a ring, and therefore extremely high heat losses occur.
2. "Pinch" effect, or the breaking of the circuit if the current exceeds a certain value, this effect being due to the mutual attraction of parallel conductors.
3. The electrical conditions of the induction furnace are unsuited to normal power-supply circuits; a bad power factor is obtained very low frequency is required, and the furnaces are usually designed to operate on single-phase supply only.

It is necessary, therefore, to use for the generation of heat a resistant material other than the metal to be melted, and the electric arc, which is a column of gas or vapor through which a current of electricity is passing, provides a ready and efficient method of generating a very intense heat in a small space.

*Paper read at a joint meeting of the Institution of Electrical Engineers and the Iron and Steel Institute, Thursday, May 8, 1919.

The most efficient method of utilizing the heat of the electric arc is obviously to make the material to be melted one pole of the circuit, or, at any rate, part of the circuit, in order that the arc may be in direct contact with the metal. The first furnace using the electric arc was designed with the container as one pole of the circuit, and a carbon electrode over the metal as the other pole of the circuit.

In order to use a higher voltage and obtain a better distribution of heat, a furnace was designed with two carbon electrodes over the charge, the current passing from one electrode to the charge, and from the charge to the other electrode, single-phase current being used, as shown diagrammatically in Fig. 1.

In order to utilize three-phase currents, a third electrode was arranged over the charge, as in Fig. 2; furnaces of this kind have been very largely adopted in connection with steel furnace practice, and still form a considerable proportion of the furnaces in operation at the present time.

Without in any way attempting to deprecate the extensive pioneer work carried out by furnaces of this type, there are several inherent disadvantages, viz.:

1. The electrode currents are not independent of each other.
2. The heat is applied to the surface of the charge only, and very little circulation of metal is caused.

With regard to (1) it will be seen that if one electrode dips into the charge, or is being driven down into the charge by automatic regulation, while the other electrodes are clear, no current passes and the electrode or electrode-operating gear is liable to be broken while a molten charge of steel takes up carbon from the electrode and upsets the analysis. Many devices have been introduced to overcome these objections, but it is well known that most users experience difficulties of this kind, and particularly so when the furnace has an acid or non-conductive lining.

With regard to (2), it will be generally acknowledged that it is more efficient to boil water by applying a gas burner below the water than on the top surface only, as very little circulation of water will result in the latter case. This applies also in the case of an electric furnace, and it is a great advantage to introduce a method of causing the metal to circulate freely.

This circulation can be effected by making the hearth of the furnace fairly thick and passing a considerable current through it by using it as one electrode; it is possible in practice to apply from 8 per cent to 10 per cent of the total energy supplied to the furnace on the hearth alone, by using a mixture of magnesite and dolomite and grading it to give the highest resistances near the charge and a negligible resistance towards the outside. Such a hearth is as strong and permanent as any basic hearth that can be made.

Some years ago, in collaboration with Mr. Etchells, the author discovered an electrical law which enables three-phase or two-phase current to be applied to a furnace with an unequal resistance in one of the phases and still maintain a truly balanced load as regards both power and power factor on the primary phases. The connections of such a furnace are shown for three-phase and two-phase working in Figs. 3 and 4, respectively, the length of O C being determined by the resistance of the furnace hearth.

In passing, it may be of interest to electrical engineers to know that an arrangement, as shown in Fig. 5, with an unequal primary star and an unequal secondary delta connected to the furnace will give a truly balanced primary load when the two electrodes are carrying equal currents, the transformer ratios varying according to the relative resistance of the furnace hearth to that of one of the arcs. The author purposely avoids giving mathematical treatment of these rather curious electrical combinations, as they would not interest many of the members.

Fig. 1.

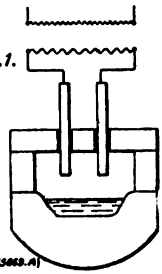


Fig. 2.

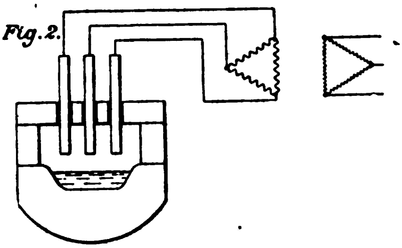


Fig. 3.

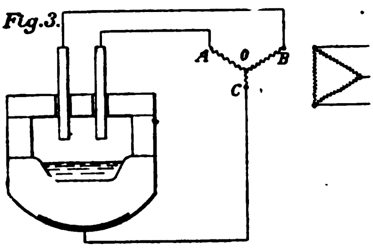


Fig. 4.

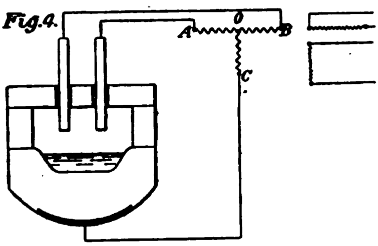


Fig. 5.

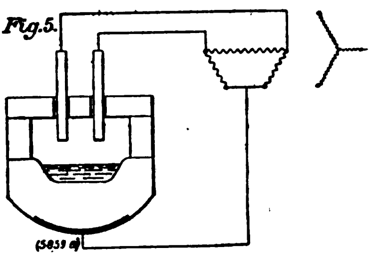


Fig. 6.

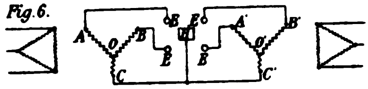


Fig. 7

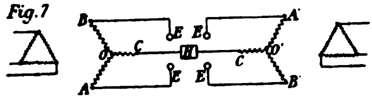
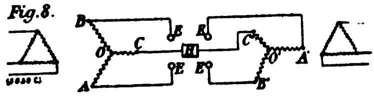


Fig. 8.



It will readily be seen that with the systems shown in Figs 3, 4 and 5, each electrode is independent, and regulation of the electrodes, either automatically or by hand, is perfectly simple, while the heat generated in the hearth of the furnace maintains the metal in constant circulation. These systems also give an extremely steady load on the primary phases, due to the buffer effect of the resistive hearth and the phase displacement caused by short-circuit currents.

Up till recently furnaces of the bottom-connection type have not been operated with an acid lining, as such lining is considered to be non-conductive, although one furnace is made with a water-cooled stud in the hearth to maintain contact with the charge. Water-cooling in the hearth of an electric furnace is surely a dangerous and inefficient makeshift, and the author is pleased to say that after some little investigation a conductive acid lining has been discovered which has operated satisfactorily for some months; this invention, full particulars of which, unfortunately, he is unable to give at present, will also solve the main difficulties in regulating top electrode furnaces when operating "acid."

A number of furnaces have been built where the full load of the transformers cannot be obtained owing to the drop in power factor with an increase of current, and it has been found in practice that the full-load current per electrode on a 50-period circuit should not exceed 8,000 ampères to 9,000 ampères.

For furnaces of five tons' capacity with transformers of 1,300 k.v.a., two-electrode furnaces are satisfactory, but above that size three or, preferably, four electrodes should be used. Figs. 6, 7 and 8 show three combinations of the transformer system illustrated in Fig. 3, E, E, E, E representing the four upper electrodes and H the hearth. In Fig. 6 the current transmitted by the hearth when the electrodes are in equal adjustment will be approximately 0.7 times the sum of the currents in the four electrodes; in Fig. 7 the current transmitted by the hearth will be nil, and in Fig. 8 it will be approximately 0.35 times the sum of the currents in the four electrodes.

It is the general practice to provide trappings on the primary windings of the transformers in order to vary the voltage, usually one voltage for melting, one for refining, and one for holding the metal at a constant temperature in the furnace; it is possible, therefore, by suitably connecting the voltage-changing switch to arrange for different degrees of bottom heating effect for the different stages of each heat, the best circulating effect being necessary of course when the metal is molten and alloy additions are being made to the bath.

There are several types of electric furnaces which operate with an independent arc, that is to say, an arc is formed between the carbon electrodes, and the charge does not form part of the circuit. These types are not efficient for steel melting, the output, energy and electrode consumptions, and lining repairs comparing very unfavorably with the direct arc furnaces.

For melting certain metals, however, such as bronze or copper, the indirect arc furnace has certain advantages, particularly when melting metals which are volatile at comparatively low temperatures. In such a case to employ a direct arc furnace would result in a very high loss due to volatilization on account of the very hot region of the arc being actually in contact with the metal.

By employing an indirect arc furnace this effect is somewhat reduced, but special means for cooling the walls of the furnace must be resorted to in order to prevent the refractories from fusing at too high a rate and thus render the cost of running prohibitive.

The reason for this is that the rays of intense heat from the arc strike directly on the walls, whereas in the direct arc furnace the intense heat is

more or less shielded by being partly buried in a crater of slag or metal, which material, in addition to acting as a shutter, also rapidly conveys the heat away.

To render the indirect arc furnace a more commercial proposition, the U. S. Bureau of Mines has adopted a method of rocking the furnace in such a manner that the molten charge is constantly washing the walls of the furnace and thus the lining is kept comparatively cool. In addition to this cooling effect, the washing of the metal has a tendency to absorb some of the metal which may have been driven off as a vapor; also the charge is kept well stirred, which results in a very uniform product.

For many kinds of work the arc furnace is quite unsuitable, such as reheating, annealing or melting of metals which have low melting temperature, and even with high melting temperature, when the quantity is small.

For such work the resistance furnace is the best type that can be employed; but where the temperature to be attained is above 1,000 degrees C., the difficulties to be met with are very numerous.

For temperatures below this figure, probably the best element is that made from an alloy of nickel chromium; but for temperatures above 1,000 dergees C. it is necessary to resort to other materials.

In the Bailey furnace a trough constructed of a highly refractory material, such as silicon-carbide, is built on the inside wall of the furnace. The trough is filled with a carbonaceous material which forms the resister. The heat from the material usually travels in an upward direction and impinges on the roof of the furnace, which acts as a reflector and reflects the heat down on to the charge. This seems a roundabout method of heating and is calculated to punish the roof material unduly. In addition to this drawback the carbon of the resister oxidizes very readily, consequently requires frequent renewal, and thus leads to the trough being gradually filled with residue or ash from the carbonaceous material, which means periodically shutting down for cleaning.

By constant effort to improve the resister type of furnace a method of construction has been brought out to overcome some of the previous difficulties and improve the efficiency of the furnace. The method consists in forming a hearth or crucible, which acts as the container for the bath of metal, the depth of the bath being relatively small compared with the surface area of the bath. The register element, or elements, if more than one is required, are arranged suspended above the bath, but as close to the metal as possible, only sufficient distance as may be required for stirring or charging being allowed. The heat generated is transmitted to the charge mainly by direct radiation.

The heating element is made of a highly refractory material, and yet not readily oxidizable at high temperatures, nor yet does the material disassociate until an exceedingly high temperature has been attained.

The element itself is built up of a number of sections dovetailed into one another and held in suspension by pressure exercised at the ends of the resister, the amount of pressure used actually being greater than that necessary to support the element; but as the resistance of the element to a certain extent depends upon the pressure, a margin of variation is allowed for.

This type of element requires no booster transformer or any extensive voltage-changing device, such as is employed in most resistive types of furnace, the amount of energy being readily varied by pressure applied to the heater unit in a manner external to the furnace.

Another feature of the furnace is that the elements are renewable even while the furnace is hot.

The construction is such as to make the furnace gas-tight, which results in an absolute minimum of zinc loss when employed on brass melting.—"Engineering."

REPAIRS TO THE *CURACA*.

An Echo of the Halifax Explosion.

In the reconstruction of the British steamer *Curaca* the Robin's Dry Dock & Repair Company has again demonstrated its ability to handle successfully difficult jobs sent to its big plant in Erie Basin, Brooklyn.

The *Curaca* was sunk during the explosion in Halifax Harbor. Her superstructure was blown away and she was buckled at about the mid-length to such an extent that when refloated the draught at buckle was 16 feet 2 inches, while the draught at bow was 7 feet and at stern 9 feet 8 inches, showing that the vessel was forced up 9 feet 2 inches from the base line forward while the stern was forced up 6 feet 6 inches.

In this condition the vessel was raised, temporarily repaired and brought to New York under her own steam, and the work of straightening and repairing commenced on April 25.

The big problem here was to restore the vessel to an even keel. On a preliminary survey this seemed to be impossible. Further examination was made and finally Mr. George J. Robinson, Vice-President of the Company, outlined a plan which subsequently proved successful. His report on the job is of more than ordinary interest and value and reads as follows:

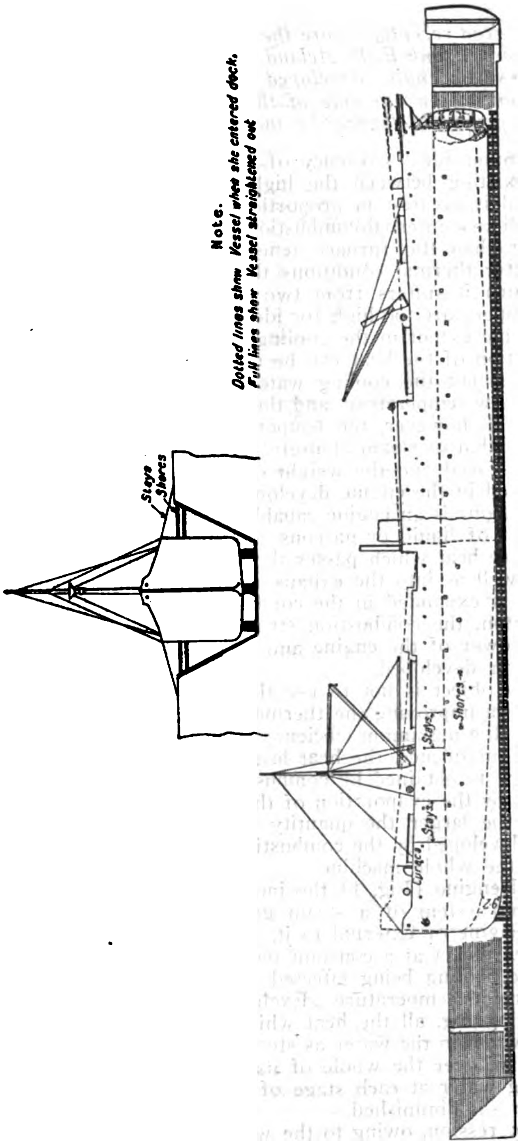
"We decided to use No. 2 graving dock for the *Curaca*. Keel blocks were set solid for a length of 36 feet, as it had not been determined exactly where the buckle was. On April 25 the vessel was warped in, placed over the blocks and plumbed up. Shores from each side of the dock were then put in place against the sides of the vessel. Wire hawsers were extended from each side of the deck and made fast ashore, so that there would be no danger of either the forward or after section listing if a break should result at the buckle point before the straightening of the vessel was completed.

The dock pumps were then started. As soon as the buckled area of the keel touched the blocks the pumps were stopped and examination made to see that the vessel was perfectly upright and over the blocks. The sides of the vessel were then gradually cut at the buckle from the deck to the water-line on both sides. The pumps were again started and as the dock was slowly freed from water the cutting was continued about 6 inches at a time.

Close observation showed that as the cutting of the plates and the pumping away of the water progressed the ends of the *Curaca* dropped gradually, being without the support of the blocks. During this operation frequent inspections were made to determine that both the forward and after sections were upright, that the hull had not twisted and that no structural change had resulted. When the dock had been pumped dry the entire keel of the *Curaca* rested on the blocks.

This operation was begun at 7:30 a.m. and was successfully completed at 3 p.m. the same day.

We found that the bottom plating around the buckle had fractured, due, probably, to the vessel having worked at this point. The bottom of the *Curaca* was also damaged by the explosion to such an extent that 54 of her bottom plates needed repairing and renewing."—"Shipbuilding and Shipping Record."



How the "Curaca" was Straightened Out.

COMBINATION INTERNAL-COMBUSTION AND STEAM ENGINE OF HIGH EFFICIENCY.

In a paper read recently before the Royal Society of Arts (England) and abstracted here, Frank E. D. Acland described a combination internal-combustion and steam engine developed by W. J. Still, in which the products of combustion act on one side of the piston and steam on the other, the steam being generated largely by the waste heat from the gas cycle.

The maximum ideal efficiency of a heat engine is obtained where the difference existing between the highest and lowest temperatures of the working fluid is greatest in proportion to the maximum temperature, and here the ordinary internal-combustion engine, with an initial temperature often higher than the furnace temperature of the boiler is capable of realizing better thermal conditions than any other form of heat engine; but in its turn it suffers from two disadvantages—it ejects its working fluid at a temperature too high for ideal conditions, and it loses heat energy to a regrettable extent in the cooling of its cylinder. In existing engines some proportion of the heat can be usefully recovered as steam from the exhaust gases, but the cooling water from the jacket is of little value, owing to its low temperature, and the efficiency of the engine itself is not augmented. If, however, the temperature of the cooling water could be maintained at that of steam at useful pressure, the efficiency of the engine would be improved and the weight of steam be usefully increased. This is accomplished in the engine developed by W. J. Still.

The Still engine is an engine capable of using in its main working cylinder, any form of liquid or gaseous fuel hitherto employed; makes use of the recoverable heat which passes through the surfaces of the combustion cylinder, as well as into the exhaust gases, for the evaporation of steam, which steam is expanded in the combustion cylinder itself on one side of the main piston, the combustion stroke acting on the other side. It increases the power of the engine and reduces the consumption of the fuel per horsepower developed.

Its primary object is not to use the waste heat for raising steam, but first to use it in improving the thermal conditions of the working cylinder, and so insure the maximum efficiency from the fuel burnt within it, diminishing, as a consequence, the heat lost in that operation. Since the maximum efficiency is obtained by combustion of the fuel in the cylinder and the minimum by the evaporation of the water in the steam generator, it is evident that the larger the quantity of steam that can be generated per horsepower developed by the combustion cycle, the lower must be the heat efficiency of the whole machine.

In the Still engine (Fig. 1) the jacket and cooling water form part of the circulating system of a steam generator, which may be an integral part of the engine or external to it. The cooling water therefore enters and leaves the jacket at a constant temperature, regulated by the pressure of steam, the cooling being effected by converting the water into steam without raising its temperature. Excluding the radiation losses, which are kept low by lagging, all the heat which passes through the walls is thus usefully recovered in the water as steam. The temperature of the cylinder wall is uniform over the whole of its exterior surface, and the heat lost to the cooling water at each stage of the cycle—compression, combustion and expansion—is diminished.

During compression, owing to the walls being at steam temperature, the incoming charge picks up heat, instead of losing it, during the greater part of the stroke, an advantage of the greatest value to the heavy-oil types of Still engines, where an air charge is taken in at the full outstroke, and is

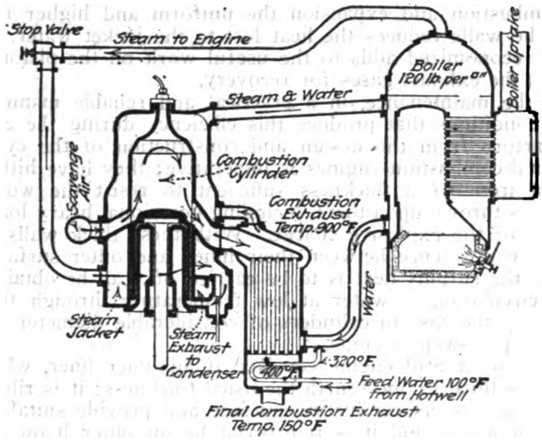


FIG. 1. CROSS-SECTION DIAGRAM OF STILL ENGINE

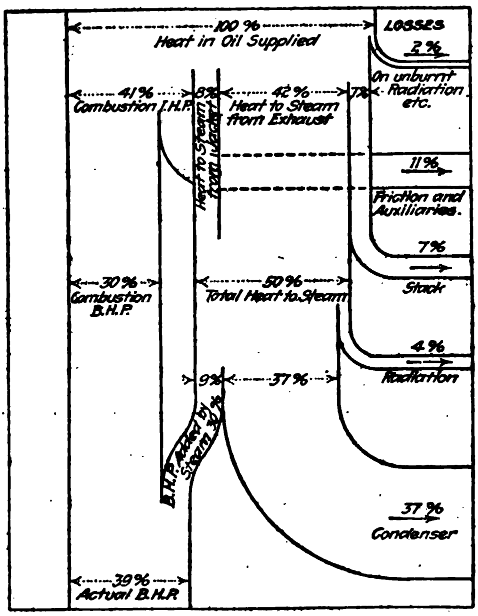


FIG. 2. CHART SHOWING DISTRIBUTION OF HEAT UNITS

compressed to a pressure where its increased temperature insures the certain ignition and combustion of the fuel that is injected into it.

During combustion and expansion the uniform and higher mean temperature of the walls reduces the heat lost to the jacket water. Some of the heat thus economized adds to the useful work on the piston, the rest passing out in the exhaust gases for recovery.

To insure the maintenance, in a practical and reliable manner, of the temperature conditions that produce this efficiency during the combustion cycle, a departure from the design and construction of the cylinders of normal internal-combustion engines is imperative; they have hitherto been made of cast iron, of a thickness sufficient to resist the working and thermal stresses thrown upon them, including occasional heavy loads caused by preignition of the explosive charge. With these thick walls or liners, the temperature difference between their inner and outer surfaces, which is essential if the surplus heat is to be carried off, can be obtained safely only by the circulation of water at low temperature through the jacket; this is especially the case in cylinders of considerable diameter or capable of high power per swept volume.

The cylinder of a Still engine consists of an inner liner, which is approximately one-third to one-fourth the usual thickness; it is ribbed externally so as to add to its conducting surface and provide suitable passage for the cooling water, and it is reinforced by an outer hoop capable of withstanding the highest pressures to be met with in working. The stresses due, in ordinary practice, to the cold water at the inlet, and the hotter water at the outlet, are suppressed in the Still cylinder, the cooling water being at a controlled and uniform temperature throughout.

STEAM FROM WASTE HEAT.

In gas and oil engines of constant-volume and constant-pressure types the combined losses in radiation—cooling water and exhaust gases—range between 75 per cent and 65 per cent. The highest indicated thermal efficiency claimed under test conditions with a Diesel engine (Mathot) 300 b.h.p., four-stroke at three-quarter load, is 47 per cent (36 per cent brake efficiency). If 4 per cent is allowed for radiation, 49 per cent of the total heat is available for recovery, and if 10 per cent efficiency is assumed for the steam cycle, the brake thermal efficiency of an engine giving this high result would be raised by 10 per cent of $49 = 4.9$; that is, $36 + 4.9 = 41$ per cent; but there is no reason why the steam generated and used under the conditions of the Still system should be limited to so low a figure. Fifteen per cent, according to the author, seems a more reasonable assumption, and even a higher figure may be anticipated, in which case a brake thermal efficiency of 44 per cent should be possible in a complete installation.

The exhaust gases take a subsidiary but important part in the cycle; their usefulness in ordinary combustion engines, in raising steam, is limited to the amount of heat recoverable between the initial temperature of the exhaust and that of, say, 50 degrees F. above the steam temperature, after which the whole volume passes away to atmosphere at a still useful temperature, less a small percentage available for feed-water heating. But in the Still engine the exhaust gases, after raising their quantum of steam, are employed in preheating all the water required for the steam generated in the jacket water and in the generator. Trials at full efficiency over long periods and steady loads, show terminal stack temperatures as low as 150 degrees F.

The piston and cylinder walls, preheated by the combustion stroke, are at a higher temperature than the steam when it is admitted and while it is being expanded; there is, therefore, no loss from condensation, and the steam exhausts at a slight superheat above the normal expansion temperature, a condition which is unattainable in any form of steam engine without direct loss of energy. With an early cut off, it can be expanded economically right out to atmospheric pressure, or below it, and be either recompressed (Stumpf cycle) or exhausted to the condenser. The steam, during expansion, forms an efficient means of cooling the piston.

The Still oil engine starts with the cylinders and pistons preheated. The air charge, from the moment of its entry into the cylinder, picks up heat from the containing walls and continues to do so during at least 70 per cent of the compression stroke, with the result that the temperature necessary for firing with certainty the first injected charge of fuel is reached with a compression pressure 50 per cent less than that required in a Diesel engine.

This fact is far reaching in its importance, for it gives to the designer great elasticity and freedom of application; for a Still heavy-oil engine can be designed for constant pressure or constant volume, or both can be employed in the same engine by correct timing of the fuel injection. It claims for its combustion cycle an efficiency higher than that of the Diesel, less weight and space per horsepower, and for its combined cycle an efficiency not less than 20 per cent higher than any prime mover which uses fuel as its source of heat.

Normal Load—The average m.e.p. from the combustion stroke was 90 pounds per square inch. The steam evaporated by the "waste heat" gave 14 pounds per square inch m.e.p. on every return stroke. This is equivalent to $90 + 28 = 118$ pounds per square inch m.e.p. in a normal four-stroke engine.

Overload—By admitting additional steam generated by fuel under the boiler, the steam m.e.p. was raised to 72 pounds per square inch; the total m.e.p. was, therefore, equal to $90 + 144 = 234$ pounds per square inch m.e.p. in a normal four-stroke engine.—"Power."

EFFECT OF WATER INJECTION ON GASOLINE ENGINES.

BUREAU OF STANDARDS TESTS SHOW NO POWER GAIN OR DECREASE OF CARBON DEPOSIT EXCEPT IN CASE OF ENGINES WITH DEFECTIVE COOLING SYSTEMS.

The practice of injecting water in conjunction with the fuel is quite common in kerosene engines, the object being to keep down the cylinder temperature under conditions of heavy loading and prevent preignition. In fact, it seems almost impossible to operate an Otto cycle kerosene engine and get satisfactory results without water injection. There also has been considerable experimentation with water injection (or induction) in gasoline engines, not by the manufacturers of the engines, but by makers of fuel conditioners and their customers, who believe that the injection of water will lessen or eliminate the formation of carbon deposit and increase the fuel economy.

In the development of aircraft engines the suggestion was made that if water injection had these effects it should be beneficial in aircraft work. The problem was assigned by the National Advisory Committee for Aeronautics to the Bureau of Standards, and an extended investigation was made. The tests were carried out on a Class B military truck engine,

and also on a Rutenber 6-cylinder, 3 x 5-in. engine, which operated at high jacket temperature. Thus, although the investigation was made at the instigation of the Aircraft Department, the tests were conducted on a truck and an automobile engine, but the results, of course, are of general application.

TWO SERIES OF TESTS MADE.

In a general way, two series of tests were made, one to determine the effect of water injection on the maximum power obtainable from an engine and its fuel economy, and the other to determine its effect upon the carbon deposit on the piston head and cylinder walls.

It may be recalled that the Class B truck engine is a 4-cylinder design with a bore of 4.75 inches and a stroke of 6 inches. The compression ratio is 3.7 and the piston displacement 425 cubic inches. Compression tests made by means of an O-Kill indicator showed an average compression of practically 48 pounds per square inch at 100 R.P.M., the water jacket temperature being 131 degrees Fahr. In the tests this engine was connected up to a 125-H.P. Sprague dynamometer, by means of which the horsepower output was measured.

In the first test the Class B engine was fitted with a Zenith L-6 carbureter. Each series of tests consisted of three runs. After the engine had been brought up to operating temperature the carbureter was adjusted for maximum engine power at full speed, only gasoline being fed into the cylinders. After the engine had been run 5 minutes under these conditions, another run was made with water injection, with the same carbureter and spark setting. Finally, during the third run, also of 5 minutes' duration, the water injection was continued, and the spark was adjusted to give the greatest engine output. Series of tests of this kind were made at speeds of 400, 600, 800, 1000 and 1200 R.P.M. The water was admitted to the manifold at a point 1¾ inches above the throttle valve, and the amount of water fed was read off from a graduated glass cylinder of 1000 cubic centimeter capacity, while the time was taken from a stopwatch. A stopcock in the line between the graduated cylinder and the intake manifold permitted of controlling the water feed.

Three complete series of tests, each covering the whole range of engine speed, were run with the Zenith carbureter. In the fourth test a Stromberg N-3 carbureter, with a 1 5/32-inch choke and a 0.0635-inch bleeder, was substituted for the Zenith, this carbureter permitting of varying the fuel mixture for different speeds. Thus, the effect of water injection could be studied when the fuel mixture was adjusted to be as lean as possible consistent with smooth running, and also when a rich mixture was being fed. In one of the runs a metal plate, with asbestos gasket, was inserted between the intake and exhaust manifold, so as to shut off some of the heat flow from the exhaust to the inlet manifold and permit of determining the effect on the operation due to the lower temperature of the mixture thus obtained.

The final tests with the Class B engine were made under spark-and-throttle conditions simulating road-operating conditions of the motor truck.

AMOUNT OF CARBON DEPOSIT.

One of the chief claims made for water injection in gasoline engines has been that it reduces the amount of carbon deposit and that it will even remove deposits already formed. In order to test the validity of this claim, a Rutenber six-cylinder, 3 x 5-inch engine was mounted on

the test stand and fitted with a fan brake to load it down. Cooling water was circulated by gravity, and in order to form a carbon deposit of some thickness the engine was run for several days with a rich mixture, late ignition, and low jacket temperature. Occasionally oil was injected into the cylinders. At the end of the 6-hour period, all interior portions were covered with a substantial layer of carbon.

The engine was then run for another 6-hour period under wide-open throttle, water being injected into the manifold as in the previous tests. During this run the waterjacket temperature was kept high, the outlet temperature being maintained constant. At the conclusion of this run the cylinder head was removed and the carbon deposit in the combustion chamber examined, but no noticeable effect was found. Other similar tests were made, the rate of water injection and the temperature of the jacket water being varied. The rate of water feed varied from 2.4 pints to 7.05 pints per hour. With the higher rates of water feed the engine power output was noticeably decreased.

The results arrived at have been summarized by the experts of the Bureau of Standards somewhat as follows:

No appreciable effect is produced upon the power, fuel economy and general operation of a gasoline engine by the injection of water into the cylinders at rates varying from 0.03 to 0.44 pound per brake horsepower-hour. When water is injected at a higher rate than 0.44 pound per brake horsepower-hour there is an appreciable decrease in the power output, fuel economy and smoothness of operation. It is quite probable that in a badly carbonized engine, or an engine of defective design, in which there are hot spots that cause preignition, the injection of the water results in an increase of power. In an engine operating at high waterjacket temperature the injection of water in amounts between 2 and 8 pounds per hour produces a softening and slight reduction of carbon, this reduction not exceeding 25 per cent and being most noticeable on the piston heads and valves. However, water injection at the maximum rate also causes a considerable reduction of power.—"Automotive Industries."

ONE THOUSAND DOLLARS WILL BE PAID FOR A SINGLE WORD.

WORLD TRADE CLUB OFFERS PRIZE FOR BEST NAME FOR "BRIT-AMS."

Can you create the one word which will best denote the United States and all parts of Britannia? If so, you will be paid at the rate of $1,000 a word. The World Trade Club of San Francisco has offered $1,000 to the person who suggests the word which, in the judgment of the club's Metric Campaign Committee, is best adapted to world-wide use.

The competition is open to all humankind. The money will be paid to the winner at noon on 15th May, 1920, by a committee appointed by President W. H. Hammer of the World Trade Club.

"Brit-Am," "Ambria," "Ambrittica," "Br-Am," "Sam-Bull" are some words thus far suggested. New names are constantly coming. The World Trade Club is offering this award because in carrying on its present campaign for the adoption of metric units by all English-speaking people—the United States, the British Isles, Canada, Australia, New Zealand, Tasmania, United South Africa and so on—it was hampered by the lack of a single short word which would express all these.

The metric units of weight and measure are now used by all the world except "Brit-Am" or "Ambrittica" or "Sam-Bull."

SOME FACTS ABOUT MONEL METAL.

By Hugh R. Williams.*

Conditions surrounding the use of high-pressure steam, particularly when superheated, are different from those met with in low or medium pressure saturated steam practice. High-pressure strains, greatly elevated temperatures and intensified erosive action, all have to be contended with, making necessary the employment of much equipment of a character different from that suitable for less severe conditions. Particularly is this so of the metals employed.

The requisites for a metal to withstand the high pressures, the total heat—possibly as high as 800 degrees F., or even higher—and the erosive action of high-pressure superheated steam are: (1) High tensile strength at all temperatures; (2) low heat conductivity; (3) high modulus of elasticity; (4) a coefficient of expansion permitting its use in combination with other essential metals; (5) capacity to resist the erosive action of the steam. If, in addition, the metal might be of a non-corroding nature, markedly resisting oxidation, it would be the ideal metal for superheated steam service.

Valve manufacturers, upon whom devolved the important necessity of producing devices to control and regulate the flow of the more powerful operating medium, were among the first to appreciate the inadequacy of their former product to handle successfully high-pressure superheated steam. Cast-steel valves were necessary to replace the extra-heavy cast-iron ones suitable for steam pressures up to 250 pounds to withstand the increased steam pressures and the much more marked elevations in temperature without unduly adding to the weight of the valves. These valves, as in the case of iron-body valves, required seat rings, valve disks and other fittings and trim made of durable and non-corroding metal to assure tight closure in operation. The fittings for iron valves, usually of brass or bronze composition, are not suitable for cast-steel valve service, for the coefficient of expansion of cast steel is nearly double that of cast iron, brass or the bronzes used for valve fittings. The difference in the expansion of the valve-body metal and the valve fittings under the range of temperature encountered in high-pressure superheated-steam service would be such as to prohibit tight closure and produce serious deformation of valve seats, etc., at unusual temperatures. The difficulty was overcome by making the fittings for cast-steel valves out of monel metal, a natural alloy of individual characteristics, possessing a coefficient of expansion practically identical to that of steel. This monel metal is non-corrodible, strong as steel, tough and ductile. It takes and retains a high finish, is a better resistant to acid and alkaline corrosion than any of the common metals and is practically impervious to oxidation. It can be machined, forged, soldered and welded, both electrically and by the oxyacetylene process.

This metal is a natural product of the distinctive nickel and copper ores mined in the Sudbury district, Ontario, Canada. The ore is a basic igneous rock carrying sulphides of nickel and copper with pyrrhotite (magnetic pyrites) as constituents—the sulphides more or less segregated—and is smelted to matte form by a special roasting process that drives off the sulphur content without disturbing the balance of the main metal ingredients, copper and nickel. The process practically eliminates the sulphur content, and the matte, refined to get rid of impurities, produces a metal of unusual uniformity of composition—nickel, 67 per cent; copper, 28 per cent; and other elements, chiefly iron from the

*The International Nickel Co.

original ore and manganese, silicon and carbon introduced during the process of refining, 5 per cent.

The physical characteristics of the metal—it was not until 1906 that a practical method of producing monel metal was developed—are decidedly distinctive and individual. They are not and cannot, so far as known, be possessed in any like measure by synthetic mixtures of similar ingredients, for no mechanical process, so far as known, can produce the intimate combination resulting from centuries of geological formature.

The requisites for a metal to withstand the severe conditions of high-pressure superheated-steam service are apparently all possessed by monel metal, as evidenced by its physical properties. The one question remaining in doubt, or not conclusively established by the physical properties of the metal under normal conditions, is its ability to withstand effectively the universally weakening effect of high temperatures.

However, a series of comprehensive and exhaustive tests conducted by some of the leading valve manufacturers showed that in this respect monel metal is quite unique among metals in having its strength less influenced by elevated temperatures than any other known metal. Under

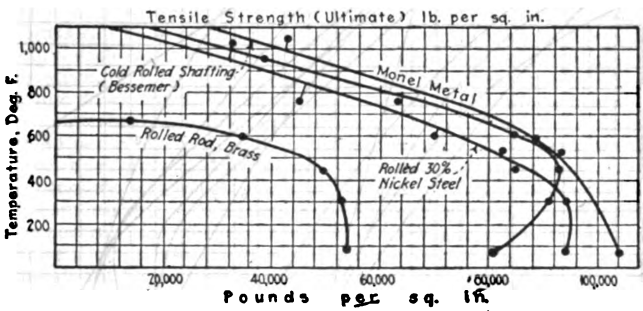


FIG. 1. BEHAVIOR OF MONEL AND NICKEL STEEL

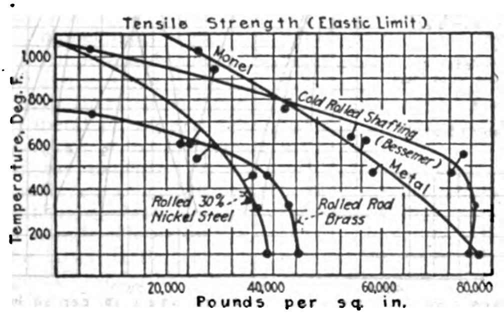


FIG. 2. ELASTIC LIMIT AT HIGH TEMPERATURES

any heat up to 1,000 degrees F., its ultimate tensile strength in rolled form is considerably greater than that of any other metal ordinarily furnished in the form of rods. It is considerably stronger than 30 per cent nickel steel at all temperatures and at high, as well as normal, temperatures is stronger than cold-rolled shafting bessemer. Even at

500 degrees F., at about which temperature cold-rolled shafting possesses its maximum strength, monel-metal rods are slightly the stronger. At the elastic limits of the various metal rods, monel metal possesses greater tensile strength than do rods of any other metal at temperatures exceeding that of about 800 degrees F. Below this temperature, bessemer cold-rolled shafting is somewhat the stronger at the elastic limits of the respective metals, but monel metal resists the action of corrosion and oxidation far more effectively.

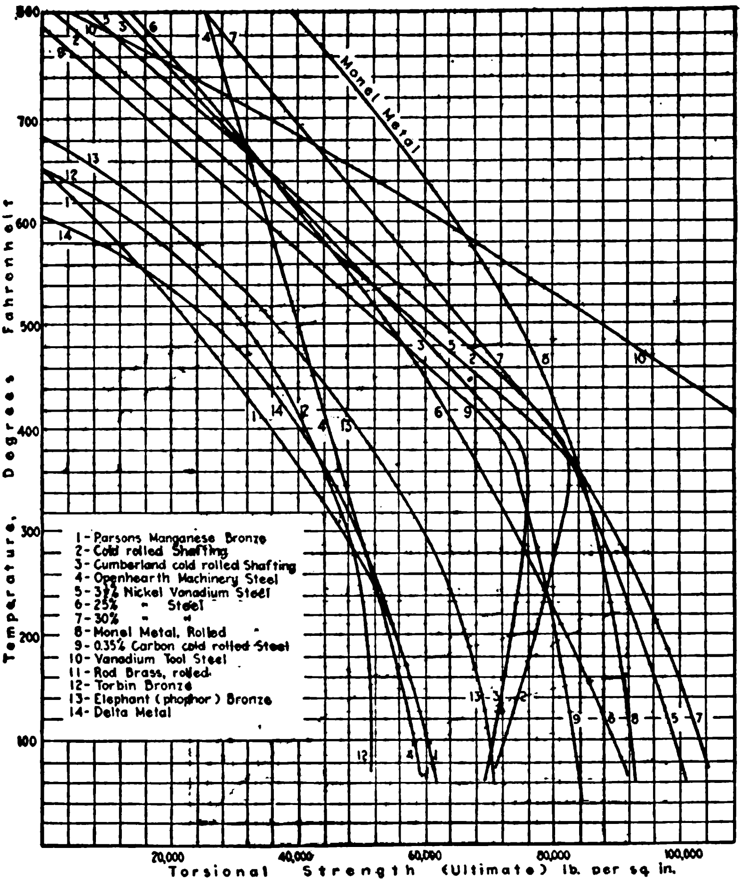


FIG. 3. BEHAVIOR OF MONEL AND OTHER METALS AT HIGH TEMPERATURES

Under torsion monel-metal rods are even stronger than bars of vanadium tool steel at temperatures in excess of 600 degrees F. and at all temperatures monel-metal rods are stronger than rods of nickel steel, any of the bronzes, cold-rolled shafting, machinery steel, delta metal, etc.

In the form of rods the metal is also of quite unusual importance in the power house. Monel-metal pump and piston rods, in particular, are

highly economical, for they are more durable than steel or bronze rods and effect a substantial saving in power consumption and the use of packing. They do not score or pit and acquire a smooth, glassy surface in service that reduces friction to a minimum, avoids wear on the packing and eliminates to a great extent the leakage evil.

For parts of valves, in fabricated forms as bolts, nuts, screws, etc., for severe service and for use in connection with cast-steel mechanisms, monel-metal rods find many uses.

Another use of monel-metal rods, in which service the metal has also assisted in power-plant practice, is in connection with steam-turbine construction. The successful development of the steam turbine has been dependent to a considerable extent upon employing a suitable metal for the blades. A metal is required that will stand up well under the erosive action of high-pressure steam, resist corrosion, will retain strength at elevated temperatures and which can be worked easily into the required shapes. The first three requirements are essential and the last a matter of practical consideration.

There are three classes of metals—brasses, nickel steel and monel metal—that meet each of the four requirements to a certain extent and on that account have been quite extensively employed for steam-turbine blading. Brass can be worked into any required shape with little difficulty, it is non-corrodible and resists the erosive action of steam fairly well, but it lacks strength for use in large units and under a considerable temperature range loses much of its strength at normal temperatures. For instance, in superheated-steam service under high pressure it may lose as much as 40 per cent, or even more, of its strength at 70 degrees F.

Nickel steel does not possess the quality to resist corrosion or to withstand high-pressure steam corrosion so desirable, if not absolutely essential, in a steam turbine. Its strength at ordinary temperatures is ample, but at a temperature in the neighborhood of 800 degrees F. it loses from 50 to 60 per cent of its normal strength.

Monel metal can be worked easily into all necessary forms for turbine blading, resists corrosion and the erosion of high-pressure steam even better than does brass, on account of its high nickel content, is very nearly as strong as nickel steel at ordinary temperatures and resists the weakening influence of elevated temperatures far more effectively. At 800 degrees F., monel metal retains as much as 66⅔ per cent of its strength at ordinary temperatures and is some 30, or more, per cent stronger than nickel steel at the same temperature.

[We believe that readers may get the impression from the foregoing, that monel metal is more widely used for turbine blades than it actually is. It does make an excellent material for such service; but most builders, because of the expense, do not use it for turbines to the extent that might be inferred from the article. Nearly all builders agree that there is slight difference between the wearing qualities of a good nickel steel and monel metal. However, from a shop viewpoint, the chief disadvantage of the latter is that it is severe on dies and increases die charges. One of the largest builders of steam turbines in this country has never used monel for the blades except for one small single-wheel impulse machine in which the steam velocities are exceedingly high. This builder doubts that monel is as good for high-speed blades as pure nickel, which is now available. Great physical strength in the high-pressure blading in modern turbines is not vital. For the sake of efficiency all such blading, whether impulse or reaction, should operate at slow speeds, which result in low mechanical stresses. For the exhaust end blading where these stresses are high low-carbon, nickel, electric-furnace steel is widely used.—Editor.]

TABLE OF PHYSICAL PROPERTIES.

Melting point of monel metal, degrees	1,360 C. (2,480 F.)
Specific gravity (cast)	8.87
Weight per cubic inch (cast), pound	0.319
Weight per cubic inch (rolled), pound	0.323
Coefficient of expansion (20 deg. C.—100 deg. C.)	0.00001375 per deg. C.
Electrical resistivity, 256 ohms per mil-foot (temp. coefficient),	0.0011 per deg. F.
Electrical conductivity	4 per cent (copper 100 per cent)
Heat conductivity	1/15 that of copper
Shrinkage	¼ in. per foot
Hardness, cast material	20-23 (Shore scleroscope)
Hardness, hot-rolled rods	27 (average shore scleroscope)
Hardness, hot-rolled rods	162 (average Brinell)
Modulus of elasticity	22,000,000-23,000,000

TESTS ON RODS.

Tensile.

Average representative tests of each of the three divisions as given:

	Yield Point, Lb. per Sq. In.	Ultimate Tensile Strength, Lb. per Sq. In.	Per Cent. Elongation in 2 In.
Up to 1 in.	63,126	94,562	40
1 1/16 in. to and including 1 11/16 in.	61,963	93,104	39
1¾ in. to and including 2 7/16 in.	50,115	87,678	42
2½ in. to and including 3½ in.	43,805	85,282	44
Over 3½ in.	47,335	84,763	43
Rectangles	56,353	85,562	42
Hexagons	60,736	87,781	40

Torsional (Average).

Shearing stress—pound per square inch on remotest fibres:

At elastic limit	31,796
At ultimate load	79,053

Compression.

Elastic limit .... 25,500 to 32,000 lb. per sq. in.

TEST ON CASTINGS.

Tensile.

Average of 172 heats tested for Isthmian Canal Commission:

Yield point	37,093 lb. per sq. in.
Tensile strength	72,281 lb. per sq. in.
Elongation in 2 inches, per cent	34
Reduction of area, per cent	32

Compression.

Elastic limit .... 12,000 to 25,500 lb. per sq. in.

—"Power."

TESTING WATERTUBE BOILERS.

The writer has recently had under his charge the testing of some eighty or ninety new watertube boilers, and the preparing of same for steaming conditions. As the method adopted has been very successful, and may also be applied to some other classes of watertube boilers, these notes may be of interest.

First, a general idea of the boilers under consideration will be given, and then the three following items discussed:

1. Character of test required.
2. Preparing for and carrying out test.
3. Preparing for steam.

The boilers referred to above were of the Yarrow type: Steam drum, 50 inches diameter by 12 feet long; water drums, 30 inches by 12 feet long; number of tubes, about 3,500 of 1-inch diameter, except the three inner rows on each side, which are 1¼ inches in diameter; average length, about 8 feet.

The steam drum carries four safety valves bolted to pads and riveted on the shells. The front head carries the main stop valve, surface blow, two water gages, three try cocks and the auxiliary feed valve. The internal auxiliary feed pipe, scum pipe and steam-dryer pipe are fitted in the drum. The manhole is 16 inches by 12 inches.

The water drums (front heads) carry the main feed valves, and one drum is fitted with a salinometer cock. Each drum has an internal feed pipe arranged in a casing so as to cause a forced circulation. Blow-down valves are fitted to the shell of each drum. The manholes are 16 inches by 12 inches. Each boiler was capable of giving 7,000 horsepower, burning oil in a closed fire-room.

CHARACTER OF TEST REQUIRED.

The character of the test required is as follows: Drums and tubes are to be tested to 400 pounds cold water pressure for one hour. The boiler, complete with all mountings, to be tested on board to 330 pounds cold water pressure for a period of six hours, a loss of eight pounds being the limit allowed in this time.

It will be readily admitted that such a test is severe and calls for careful preparation.

PREPARING FOR AND CARRYING OUT TEST.

As they come from the shop the boilers are usually in a very dirty condition; oil and grease from the tube expanders, chips of metal, scale, rivet heads, etc., are sure to be found in plenty. The oil need not trouble us at this stage, but all particles of metal, or anything liable to damage the valves, must be thoroughly blown out with an air hose; also each individual tube should be blown in order to ensure that it is clear. The boiler valves, though new, should all be taken apart, cleaned, and given a rub with finest emery to ensure a perfect seat. It is impossible to lay too much stress on the fact that this work must be faultless; a drop, however slight, is enough to spoil a perfect test.

After all the valves have been attended to, the safety valves may be gagged, the manhole doors carefully put on, air cock opened and the boiler filled slowly. For obtaining the best results all the air should be driven out, and this is sometimes a difficult matter. The air cock is usually fitted to the top of the drum, but a certain amount of air is trapped in the safety valves and in the main stop if this rises much

above the level of the top of the drum. However, the amount trapped is small, and it is preferable to allow it to remain rather than to open the valve, as there is sure to be some little particle of grit in the water that may lodge on the seat and necessitate regrinding.

Having the boiler full, we may proceed to pump up by the hand pump to about 100 pounds. At this pressure it is well to make a thorough examination of the boiler, looking to all seams, rivets, mountings and tubes. Any leaky tubes can be easily detected by a long lighted taper passed between the rows. Assuming all is well, the pressure may be increased to 200 pounds, and later to 250 pounds, a good examination being made at each pressure. The manhole doors will probably require a little tightening up by this time.

If any fittings should be found porous they must be dealt with as experience dictates. A very small leak may be stopped by peening with a hammer or by a calking tool, care being taken to relieve the pressure first. If a casting is spongy it should be removed and a new one fitted in place. Valves not exposed to steam, as, for instance, the main feeds, may be tinned, if necessary.

Let us now assume that all is satisfactory and that the pressure on the boiler has been raised to 330 pounds without any leaks. We can relieve the pressure and prepare the boiler room for the final test.

All gages to be used should be calibrated and large gages should be used, so that a drop of even half a pound may be noted. Boiler room hatches should be closed or covered with canvas, and all openings in bulkheads plugged up, in order that the boiler may not be subjected to any draft. Stacks must also be covered, and all openings to the furnaces closed.

When this is done, and the necessary thermometers, etc., are in place, the inspector may be called, the boilers pumped up to the full pressure, and the hand pump locked or disconnected.

Readings should be taken every half hour. If it is desired to gage any part of the boiler while under test, suitable arrangements must be made beforehand.

The writer took some measurements of the boilers both before the pressure was applied, and while it was on, with the following results: After a 400-pound test on the shell and tubes (all openings for fittings being blanked) the part faced for the main stop showed a permanent distortion of thirty-seven thousandths, and the faces for water gage fittings as much as twenty-five thousandths. All these faces had to be scraped up again before fitting the mountings. Before applying the 330-pound test with mountings in place, measurements were taken between port and the starboard water gage fittings (top) and found to be 24 inches. While the pressure was on, this showed 24¼ inches. The fittings were bolted to extensions on the head about 8 inches long. A tram was also made from the boiler shell to the top of the main stop spindle, and under this pressure showed a difference of ⅛ inch. The head was ¾ inch thick.

PREPARING FOR STEAM.

After the test has been passed, we may proceed to clean the boilers, as follows: Having emptied them and removed the bottom manhole doors, we put into each bottom drum 25 pounds of soda, 6 pounds of lye and 5 gallons of coal oil. Replace the manholes and put in enough water to fill the bottom drums. Then connect a temporary steam pipe to the bottom blows and admit steam carefully at about 40 pounds pressure. The steam will condense, and, in time, bring the water to the boiling point.

When the pressure on the boiler is approaching that of the incoming steam, the surface blow may be slightly opened to maintain a pressure of 25 or 30 pounds. The condensation of steam will soon fill the boiler to the level of the scum pan, and all oil and grease will be removed in this manner.

Boiling is continued about 70 hours. At the end of this time, the boilers may be emptied and allowed to cool and then the manual part of the cleaning will commence. The tools required are few and simple: Wire hand brushes, tube brushes, wire cables fitted with sockets into which the tube brushes are screwed, scrapers, and a bale of rags, are all that we require.

First the internal pipes are removed and the top drum brushed down thoroughly with the wire brushes, care being taken to clean well around the tube ends. Sometimes there is a little mill scale found on the heads, especially around the flange, and this can be removed with scrapers. It is not well to try to chip it, as the head must be left as smooth as possible.

When this is finished, the tubes may be started. The wire brushes should be wrapped with rags, as this leaves the tubes in a fine polished condition, whereas a wire brush alone makes them look streaky. The brushes will only clean the parallel part of the tubes, and the bell-mouth must be "poked" by means of a stick wrapped with rags.

After this is done, the drum and tubes may be blown out with the air hose, and then the bottom ends of the tubes "poked" in a similar manner and the bottom drums wire-brushed, and, if necessary, washed with coal oil.

As soon as the bottom drums are finished and attention has been given to all the small outlets for water gage fittings, etc., which are easily choked up, the boiler may be inspected by the chief engineer of the ship, who should, in the case of straight-tube boilers, "sight" every tube from the top drum, while his assistant holds a light at the bottom. If the inspection is satisfactory, the internal pipes may be cleaned and replaced, and the boiler filled to the working level with distilled water.

Should the boilers not be required for any considerable time, it is better to fill them right up to the air cock with slightly alkaline water, or they may be left dry and a tray of burning charcoal introduced into each bottom drum and the boiler closed entirely.—"International Marine Engineering."

HEAT-TREATMENT OF LOW-CARBON STEEL.*

By W. M. Wilkie.

In annealing, the operation consists in (1) slowly raising the steel to its upper critical temperature; (2) holding it or soaking it at this temperature for a certain time, depending on the size of piece being treated and the condition of previous treatment; and (3) cooling from annealing to atmospheric temperature.

Theoretically, as the steel is heated to the lower critical point, the usual transformation from the phases of softened steel to austenite is commenced, and continues through the critical range, so that when the upper critical point is reached the steel should be all austenite. As a result of this transformation all pre-existing structures, no matter how coarse, are obliterated, and the mass assumes an amorphous structure, providing sufficient time has been taken in heating to permit these changes to occur.

*Abstracted from the "Journal of the American Society of Mechanical Engineers."

Further, if the temperature is carried above the upper critical point the tendency is for this amorphous structure to become crystalline again. Hence, in annealing steel its temperature should theoretically be raised to the upper critical point only, but practically there are conditions that modify this. For example, in cast steels we find that temperatures considerably higher (as much as 180 degrees F. above the theoretical temperatures), combined with long soaking, are required to break up the coarse-grained structure.

TABLE I.—ANNEALING TEMPERATURES FOR STEEL.

Range of Carbon Content, per cent.	Range of Annealing Temperatures, F.
Less than 0.12	1607 to 1697
0.12 to 0.25	1544 " 1598
0.30 " 0.49	1499 " 1544
0.50 " 1.00	1454 " 1499

Mechanical work on steel seems to have the same effect—namely, it breaks up this intercrystalline film, and as a result we find that the more a piece of steel has been worked, the closer the annealing temperatures can be kept to the upper critical point, while the nearer the steel is to the unworked or cast condition the higher the temperatures must be. The ranges of temperatures given in Table I have been recommended by the Committee on Heat-treatment of the American Society for Testing Materials.

The time of soaking depends largely upon the size of the object being annealed, but sufficient time should be given to permit the object to be heated throughout to the annealing temperature and allow the internal changes to occur. Pieces 12 inches thick require about one hour.

The rate of cooling after annealing is controlled by the product required. If a maximum softness and ductility is desired, for instance, for easy machining, we cool as slowly as possible by either burying the article in a bed of insulating material or allowing it to cool in the furnace. If, however, we wish to retain a certain amount of hardness and strength we must cool faster, as in air, or, if a low-carbon steel, in water; but to do so we must sacrifice a certain amount of ductility and softness. As a general rule, the lower the carbon the more rapid may be the cooling. For example—

1. Steels with carbon content below 0.15 per cent can be quenched in water and remain ductile, and at the same time have maximum hardness and strength.
2. Steels with carbon content below 0.30 per cent can be quenched in oil with results similar to 0.15 per cent carbon quenched in water.
3. Steels with carbon above 0.30 per cent cannot be cooled so rapidly without destroying the ductility of the steel, and so must be cooled in the furnace, or in the air when extreme softness is not required.

The size of objects must also be considered along with the product desired. Other things being equal, the larger the object the more quickly must it be cooled, as long as the cooling is not so rapid as to set up strains.

As an example of the effect of rate of cooling on the strength of steel, we have a system of strengthening steels on high-explosive shells known as the Sandberg air-cooling method. By this method air at pressures ranging from 6 ounces to 15 ounces, usually around 10 ounces, is either passed around the shell-forging or forced to impinge on the revolving shell in such a way that a shell which would normally cool in 30 minutes

or more in the open air, is cooled in 6 to 8 minutes. As a result the ultimate strength is raised with an increase in the elastic ratio, but with little variation in the ductility of the steel. This process is really an intermediate treatment between annealing and the regular hardening of steel, due to the formation of sorbite. Another way to get this sorbitic steel is first to quench steel from the upper critical range, thus retaining its fine-grained structure, but leaving the steel very hard and without ductility. The steel is now reheated close to but below the lower critical range—say, from 932 degrees to 1202 degrees F., when it loses its hardness and becomes ductile. The reason for this is that the steel has been changed from martensite to sorbite. This is really a hardening and tempering treatment, but is referred to here because it is sometimes called a double annealing treatment.

The hardening temperatures in Table II are given by E. F. Houghton and Co.

TABLE II.—RELATION OF HARDENING TEMPERATURE TO CARBON CONTENT.

Carbon Content, per cent.	Hardening Temperature, F.
Up to 0.20	1600 to 1650
0.20 to 0.35	1550 " 1600
0.35 " 0.50	1500 " 1550
0.50 " 0.70	1450 " 1500
0.70 " 0.90	1400 " 1450
0.90 or over	1350 " 1400

Steel is most commonly quenched in oil or water. In either case care must be taken, especially with high-carbon steels, to prevent cracking as the result of strains set up by too rapid cooling. If oil or water is used, the object should be withdrawn from the bath before its temperature has fallen below 212 degrees F., and the drawing or tempering treatment should follow immediately.

If a piece of polished steel is heated in an ordinary furnace a thin film of oxides will form on its surface. The colors of this film change with temperature, and so in tempering they are generally used as an indication of the temperature of the steel. The steel should have at least one polished face so that this film may be seen.

An alternative method to the determination of temper by color is to temper by heating in an oil or salt bath. Oilbaths can be used up to temperatures of 500 degrees F.; above this, fused-salt baths are required. The article to be tempered is put into the bath, brought up to and held at the required temperature for a certain length of time, and then cooled, either rapidly or slowly. This takes longer than the color method, but with low temperatures the results are more satisfactory, because the temperature of the bath can be controlled with a pyrometer. The tempering temperatures given in Table III are taken from a handbook issued by the Midvale Steel Company.

TABLE III.—TEMPERING TEMPERATURES FOR STEELS.

Temperature for 1 Hour F.	Color.	Temperature for 8 mins. F.	Uses.
370	Faint yellow	460	Scrapers, brass-turning tools, reamers, taps, milling-cutters, saw-teeth.
390	Light straw	510	Twist-drills, lathe-tools, planer-tools, finishing-tools.

410	Dark straw	560	Stone-tools, hammer-faces, chisels for hard work, boring-cutters.
430	Brown	610	Trephining tools, stamps.
450	Purple	640	Cold-chisels for ordinary work, carpenters' tools, picks, cold-punches, shear-blades, slicing-tools, slotter-tools.
490	Dark blue	660	Hot-chisels, tools for hot work, springs.
510	Light blue	710	Springs, screw-drivers.

Two sets of temperatures are shown, one being specified for a time interval of 8 minutes and the other for 1 hour. For the finest work the longer time is preferable, while for ordinary rough work 8 minutes is sufficient after the steel has reached the specified temperature. The rate of cooling after tempering seems to be immaterial, and the piece can be cooled at any rate, providing that in large pieces it is sufficiently slow to prevent strains.

How are we to know if we have given a piece of steel the very best possible treatment? The best method is by microscopic examination of polished and etched sections, but this requires a certain expense for laboratory equipment and upkeep, which may prevent an ordinary commercial plant from attempting such a refinement. However, I would certainly recommend any firm that has any large amount of heat-treatment to do to install such an equipment, which can be purchased for 250 to 500 dollars. Its intelligent use will save its cost in a short time.

The other method is by examination of fractures of small test-bars. Steel heated to correct temperatures will show the finest possible grain, whereas underheated steel has not had its grain structure refined sufficiently, and so will not be at its best. On the other hand, overheated steel will have a coarser structure, depending on the extent of overheating.

To determine the proper quenching temperature of any particular grade of steel it is only necessary to heat pieces to various temperatures not more than 36 degrees F. apart, quench in water, break them, and examine the fractures. The temperature producing the finest grain should be used for annealing and hardening.

Similarly, to determine tempering temperatures, several pieces should be hardened, then tempered to various degrees and cooled in air. Samples, say six, reheated to temperatures varying by 180 degrees, from 570 degrees to 1470 degrees F., will show a considerable range of properties, and the drawing temperature of the piece giving the desired results can be used.

The following precautions should be observed in the heat-treatment of steel:

1. Do not put a cold piece of steel into a highly heated furnace. Either reheat it or put it into a cold furnace and allow it to heat with the furnace. This precaution is especially applicable in cold weather. Remember also that the changes occurring in the critical range are not instantaneous. The steel must be given time to change. It does not pay to rush the heating. Raise the temperature slowly.

2. Allow the piece to soak at the quenching temperature until it is uniformly heated throughout, the length of time depending on the size of the piece and the rate of heating.

3. Do not allow the piece to be directly on the hearth of the furnace, but have it supported at a sufficient height for free circulation of gases on all sides. Long pieces should have sufficient support to prevent sagging.

4. Never allow a piece, especially if it has points or sharp corners, to come in contact with flame.

5. In quenching, immerse the piece with the axis vertical. This will prevent excessive warping or cracks due to unequal contraction in cooling.

6. Care should be taken that no sharp grooves, corners, or seams, which may develop into cracks, are left on pieces to be quenched.

7. In drawing the temper of a large piece in a furnace, never put it in a furnace hotter than the drawing temperature, nor allow the furnace to exceed the required temperature, otherwise the steel will be softer than required.

8. In drawing large pieces, soak a sufficient time at the desired drawing temperature to allow the heat to affect the center of the piece.

9. In annealing and quenching, never depend on the eye to judge the temperature; use a pyrometer.—"Mechanical World."

THE IMPORTANCE OF ENGINEERING TO THE NAVY.

EXTRACTS FROM AN ADDRESS DELIVERED BY HON. JOSEPHUS DANIELS, SECRETARY OF THE NAVY, AT THE OPENING EXERCISES OF THE POST GRADUATE SCHOOL, AT ANNAPOLIS, THURSDAY NIGHT, JUNE 12TH.

Engineering has come to be the chief profession in America, for everything that moves depends upon the genius, knowledge and skill of men who are as much at home in overalls as in dinner coats, who love to make the wheels go round, and who understand that motive power is the center of world progress and world prosperity.

There are four callings that add most to the sum of human wealth—the farmer, the miner, the artisan and the engineer. The last is an old profession, as the water system and baths and roads of ancient Rome testify, but its expansion to cover all fields has been the outstanding creator of wealth of our day along new lines. It makes land, the one thing it was said man could not create. It carried the water power of Niagaras, great and small, to distant centers. It increases the fruit and grain of the soil and multiplies the product of the artisan and gives new value to mineral deposits. It makes possible girdling the globe on the sea, under the sea and in the air. And yet, because it works in the bowels of the earth and in the lowest decks of the ship and in places away from the crowd, the recognition it deserves has not hitherto been given to these creators of wealth.

When one takes a voyage across the ocean (and more Americans have crossed this year than in the preceding hundred years) he understands as never before what he owes to the men in the fire-rooms, far below the water's edge, who in a temperature that compels return to Adam's style of garment, make possible safe transportation in comfort. A visit from the bridge to the engine-rooms is an education in the achievement of the engineer, for in our modern use of the term a naval constructor as well as the men who design and build and operate the great engines in a modern ship is an engineer. The captain presses a button, and lo, the ship reduces or increases its speed, and as you see this modern miracle you almost imagine it is automatic. But you leave the bridge and go below if you can squeeze into narrow compartments where men spend their lives and climb down narrow iron steps. The temperature rises and by the time you reach the fire-room you wish you had left your coat and vest and shirt on deck, for you find the men working in a heat they cannot endure longer than four hours. As you stand there, streaming with perspiration, you take off your hats in honor to the men who in grime and heat are carrying you to your destination.

When you are again on deck, enjoying the cool breeze or partaking of a good meal, you cannot forget that below decks toiling in the grim unlighted places are the engineering force of the ship. No man in the world lives with the comforts of life without obligation to other men who sweat and strive to give him the comforts and he seldom stops to ask to whom he is indebted.

I never felt so fully the obligation to these men who keep the engines going as upon a recent trip across on the *Leviathan* and back on the *Mt. Vernon*. Heaving coal into the furnace is not my long suit, but I tried a turn at it along with the boys, and was glad it was only minutes instead of hours that held me to the hard task with the thermometer at 135 degrees. Thanks to American engineering skill, we run both these former German ships on less coal than when the much vaunted German efficients operated them, and we give better accommodations and are arranging for otherwise improving conditions aboard ship for these essential men. There is no more inspiring story of the war than the heroism of the men on the *Mt. Vernon* when she was torpedoed. This characterized all the men of every rating from Capt. Dismukes, a skilled engineer officer, to the youngest recruit. It was collective heroism. But it was conspicuously true in the fire-room where the torpedo struck and where thirty-seven men were killed at the post of duty in the darkness without a moment's warning. Immediately after the torpedo struck there was not only a maelstrom of inrushing water, but the air was filled with soot and cinders and the shovels in the fire-room were thrown in various directions. The lights were put out and the fire-rooms were in darkness, but every man stuck to his job, the reserve crew hastened to lend assistance, climbing into what appeared to be a regular death-trap, and when it almost appeared to every one in those fire-rooms that the ship was sinking. And yet by collective heroism, under plans already worked out by the captain and officers, mainly due to the engineering force, the miracle of that ship returning to port under its own power was witnessed to the glory of the American Navy. Individual heroism entitles Chief Water Tender O'Connor and Patrick Fitzgerald to rank with immortals; Hoke Smith to a high place; distinction to Commander Guttormsen, chief engineer, and the whole engineering force; all honor Executive Officer Commander Staton and Lieut. Doyle, with the other officers and men who made up Capt. Dismukes' minute-men.

Before this opportunity to display heroism, engineering skill had made possible in an incredibly short time the repair of the destruction the German had wrought before leaving this and other ships—one of the most subtle achievements of the war. But it was only a signal instance of engineering skill and illustrative of what was done in designing and building submarine chasers, mine-layers, Eagles, destroyers and other craft the character of the war demanded. The flight of the NC across the Atlantic was due to engineering skill and vision, as was the adoption of the electric drive on dreadnaughts.

We have made progress in naval engineering, but what has been done is but an earnest of what the younger men in their day will achieve unless we are to mark time and surrender leadership to others. In the immediate expansion of the merchant marine and in the larger ships the Navy is building, there will be not only ample opportunity, but compulsion for greater study of engineering progress afloat than in any previous decade. With automobiles and air-craft making new records on the land and in the air, engineering skill must be applied to faster ships and better working conditions on naval craft and on merchant vessels.

There must be specialization, but in addition there must be more general study of engineering problems by all the officers in the Navy. It

ought to be impossible for any naval officers to reach the grade of captain who had not had an actual tour of duty as engineer officer as well to have studied engineering in the Academy and particularly in this post-graduate school.

Although it is now a generally recognized fact, yet it is well to emphasize that the policy which was established some years ago of requiring all officers of the Navy to be educated as engineers and line officers and to require all officers to be capable of performing engineering duties has done more to improve the general efficiency of the Navy than any other one thing.

This is due to the fact that such a large part of every officer's duties involves handling or operating some kind of an engine or machine which is used for a fighting purpose in ships of the Navy and it is of course evident that the best results can only be obtained when the officers handling these machines have a thorough understanding of their mechanics and their methods of operation.

It is frequently argued that the field of engineering is so vast that no one person can properly master it and in addition be a master of the other branches of the Naval profession. It is true that the field of engineers is a vast one because in our ships of war we utilize machines for military purposes which involves nearly all the branches of this wonderful science. We have all the various types of steam engines, and, we might well say, *gun engines,* but the point which is frequently overlooked, is, that there is a vast difference between the knowledge required to operate a machine and to build one. One does not need to know how to build an automobile engine in order that he may operate it in driving the automobile, although he can operate his automobile with better success and efficiency if he has some knowledge of the principles which underlie the construction of the engine, and it is due to this fact that it has been demonstrated that the officers of the Navy as a whole, can be trained to operate the engines and to do the general engineering duties, although it may not be practicable to train all officers to be expert designers of the engines. A knowledge of one kind of machine also greatly assists one in understanding another kind of machine and if an officer has experience in operating the steam and electrical engines of a ship, he is better fitted to operate the gun engines of the ship and to operate the ship as a whole, which is really one large machine. The operation of the engineering departments of our ship also involves a large amount of executive work, due to the large number of men required in the engineering departments and the experience which officers doing engineering duty have of this executive nature also fits them for the other executive duties of the ship, including that of the commanding officer.

An officer's entire service in the Navy is one of continuous training and should be one of progressive development, looking to the day when he is placed in command of the largest type of fighting ship, and if an officer has served in each of the various departments of a ship's organization, particularly in the engineering department, he is thereby much more capable to exercise the high office of command and is thereby better fitted to train and prepare his ship as a whole to meet the acid test of the day of battle.

Recognizing, however, that there is a particular need for a small number of officers who have a more expert knowledge than can be required of all officers of the various types of the engines and machines, including the guns, which are placed in our fighting ships, we have established this special school, called the Post Graduate Department of the Naval Academy, to give a comparatively small number of officers of special qualifications this opportunity for specialization. It is for the

purpose of this special training as experts that you gentlemen are assigned to this school. The number of officers needed for this purpose, however, is not large and it is not deemed desirable now for a large and separately organized corps to perform these duties, although there must be a few officers who will be especially designated for the service as expert engineers throughout the entire length of their service in the Navy.

The time spent in gaining this expert knowledge is not lost, but is of the greater value, even though an officer may not be designated for this special duty throughout his career. On the other hand, he will be much better fitted to perform any of the various duties which he may have, due to this special training, for engineering proficiency is the most needed of all efficiency in naval officers.

I urge you to use every endeavor to utilize this opportunity to develop your expert knowledge, with the idea principally in mind, that you will thereby be a more efficient officer and of greater value to the Navy and to your country for having taken this course, even though you may not be assigned the duties of an expert in any particular work, because after all, the highest ambition is to be a hundred per cent good, all-around officer of the Navy, ready to perform any duty which may be assigned to you.

Due to various reasons, it has appeared to me, sufficient importance is not given by the officers of the Navy, as a whole, to the value of engineering duties, and I think it would be well if all young officers at a suitable time shortly after completion of their course at the Naval Academy were required to perform engineering duties in a ship for a certain continuous and definite time, as it is my opinion that this experience would serve as the best foundation for all the duties which the young officer would be called upon to perform in his future service.

It should be realized that the officer or man engaged in running the engines and boilers of a ship is doing just as much to defeat the enemy as the officer or a man who is operating the guns and turrets of the ship. Their skill, both mechanical and personal, is utilized for one military purpose, that of fighting the ship. Without practical engineering and practical ordnance experience the man on the bridge cannot be as efficient as with this practical knowledge. The more he knows about the driving power of his ship from experience below, the better is he fitted to command a ship or a division or a fleet, and the better qualified he will be in tactic and in strategy. It is a false theory that some men shall give themselves to problems of battle practice, others to engineering and to ordnance. The great commander is greater if he has been trained in all, for knowledge of engineer and ordnance duties is essential to the highest when in command in modern machine warfare.

The Navy is first of all a great practical University, with schools for teaching all the way from the three R's to the highest naval strategy. Our greatest officers are students and teachers. Every man in the Navy is a student. When he quits studying he ought to quit the Navy. When Admiral Sims concluded his great work abroad, he thought—and who will deny he was right—that the highest duty he could perform to increase the initiative and efficiency of the Navy was as President of the War College, the head of our great university and extensive educational system. The future of the Navy depends upon the vision to see and translate the larger problems of naval equipment and naval strategy. Here in the Post Graduate School, at the War College, at the Naval Academy, at all the schools is the real motive power which must drive the Navy upon larger seas and for broader usefulness.

In considering the broad field of Engineering and the accomplishments

of the Navy in this science, it is inspiring to know that one of the greatest achievements of our Navy during the past half century is the adaption of electric propelling machinery for our huge battleships. Too much praise for this advance cannot be given to Rear Admiral R. S. Griffin, Chief of the Bureau of Steam Engineering, to Mr. W. L. R. Emmett of the General Electric Company, and to the other engineers in civil life who have joined our naval officers in bringing this unique and far-reaching accomplishment to a successful issue.

The suitability of the electric drive for naval vessels was under discussion by our experts for several years: it is first tried out on the collier *Jupiter,* where it met every test as to economy, facility of operation and endurance. As a result of the very satisfactory service of the *Jupiter,* it was decided to adopt this type of machinery for the battleship *New Mexico,* and her successful operation has given great gratification to every one concerned since these results are a notable monument to American progress in the field of engineering.

The electric drive meets, in a peculiar way, the many requirements of battleship propulsion. Generally speaking, it is more economical, with a resultant saving of a large amount of fuel, thus making the battleship a more effective fighting machine, as it gives her a larger radius of action. It is also capable of withstanding long and hard service, and the cost of repair and upkeep is much less than that of any other type.

With the electric drive, the battleship can also be given an enormously greater protection against torpedoes and mines. This follows, since the various units of the machinery can be isolated in watertight compartments so the one or more torpedo explosions might occur in the machinery spaces, and such explosions would not only fail to sink the ship, but she would be able to continue at a very high rate of speed. For instance, one of the two main turbo-generators could be destroyed by a torpedo and the compartment flooded, and yet the ship could continue to run at about three-quarter of her maximum speed. This, for a fighting ship, is an advantage of the greatest value.

Again, the electric type of propelling machinery is peculiarly flexible in its operation, so that the ship can be driven at low as well as maximum speeds, with very nearly the same relative percentages of economy. This flexibility arises from the fact that changes can be made in electric circuits almost in the twinkling of an eye, while other types of propelling machinery must be designed to obtain the best economies at some particular speed, or else a compromise must be made to meet the extremes of high and low speeds.

The adoption of the electric drive is also a notable indication of the knowledge of engineering science which now exists generally among the officers of the Navy, and is as well a tribute to the high training and marked skill of the civilian engineers and naval experts who brought this epoch-making design to a successful finish.

This is peculiarly true, since it was this expert knowledge which prevented the opposition of some eminent civilian engineers against such a radical departure in battleship machinery from blocking the installation of the electric drive in the *New Mexico.* And, it served also as a bulwark against the business interests of those shipbuilders who, for reasons of their own, opposed this far-reaching development.

It is true that there were some officers of the Navy who hesitated in advocating so revolutionary an advance, fearing that it would be perhaps a hazardous experiment to make on so large a scale in a capital ship. Some thought that there might be serious danger in using the large voltages which would be necessary, and which might in some way be a menace to the ship.

All such fears have proven groundless, however, and our pride may well be stirred by the fact that the United States Navy was the first in the world to drive a battleship by electrical power and to do this with full success in every way.

In all this, we have but won another in the long list of triumphs which have given American engineers so high a place among the world's leaders in invention and mechanical progress—those pioneers whose work, in applying nature's resources to the use of man, has been of such inestimable service to our race.

SCAVENGING IN TWO-STROKE-CYCLE DIESEL ENGINES.

By Nicholas M. Trapnell.

The purpose of scavenging is to clear the cylinder of burnt gases at the end of the power stroke and to fill it with fresh air ready for compression. This article describes methods of scavenging and types of scavenging pumps.

Scavenging in the two-stroke-cycle Diesel engine is accomplished by admitting air, at a pressure of from 3 to 6 pounds per square inch, to the

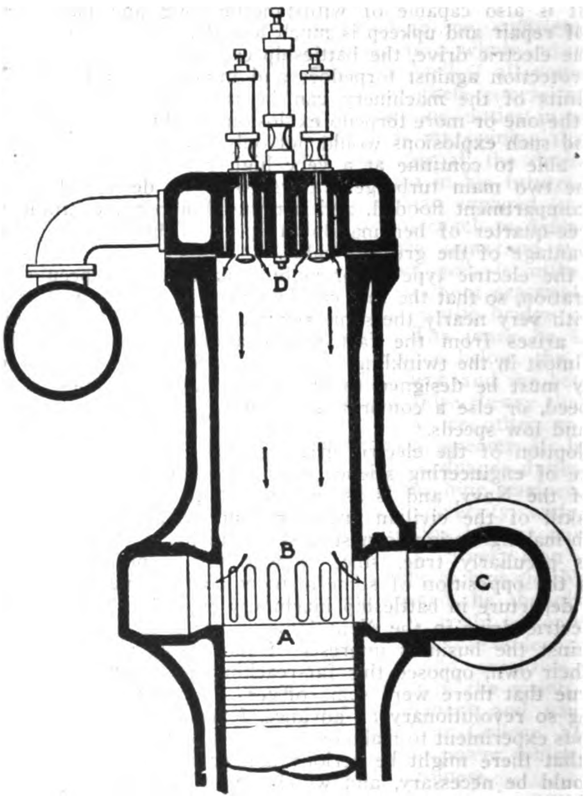


Fig. 1. Scavenging by Valves in Cylinder Head.

cylinder in such a manner that it forces the products of combustion out through the exhaust ports and fills the cylinder with air ready to be compressed on the next stroke of the piston. There are several methods of admitting the air to the cylinder. The first method used was by means of valves in the cylinder head, as shown in Fig. 1. As the piston A nears the end of the power stroke it uncovers the exhaust ports B in the bottom of the cylinder, which allows the exhaust gases to blow out into the exhaust pipe C. Shortly after the piston begins to uncover the exhaust ports, the scavenging valves D in the cylinder head open and allow the scavenging air to sweep down, forcing the products of combustion out through the exhaust ports and filling the cylinder with fresh air. The piston then starts on the return stroke and closes the exhaust ports after which the scavenging valves close and compression begins.

The scavenging valves are usually set to open some time after the piston has begun to uncover the exhaust ports, so as to allow the pressure in the cylinder to drop nearly to that of the atmosphere before the valves open; otherwise there is danger that the exhaust gases will back up through the scavenging valves into the scavenging-air receiver. The valves are usually set to close shortly after the piston has closed the exhaust ports, thereby allowing the pressure in the cylinder to build up to that of the scavenging-air receiver and giving a slightly higher pressure at the end

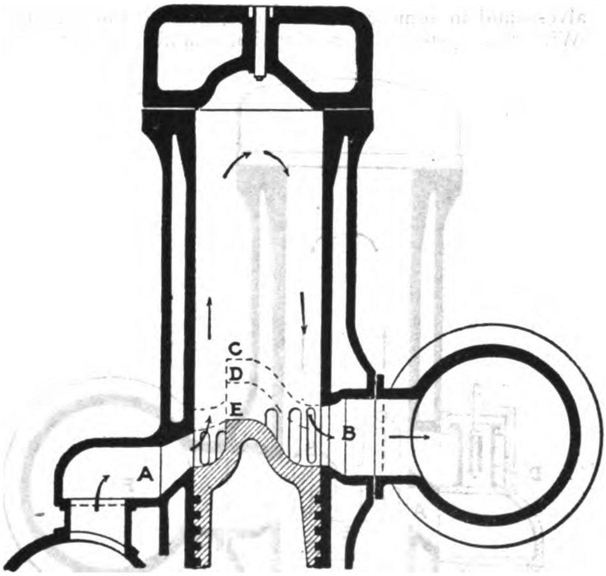


FIG. 2. SCAVENGING BY PORTS IN CYLINDER WALL.

of the compression stroke than would be the case if the scavenging valves closed as soon as the piston closed the exhaust ports.

This method of scavenging is very rapid and efficient and is extensively used in the smaller engines; but, as a large valve opening is necessary to admit the required amount of air to the cylinder, usually two and often four valves per cylinder are necessary in the larger engines. This leads to excessive complication of the valve gear, especially in large engines,

where six valves per cylinder would be necessary, including the fuel-inlet and starting valves. Also, the excessive number of valves greatly weakens the cylinder head, which should be pierced by as few openings as possible because of the high pressures and temperatures reached in Diesel-engine cylinders.

To overcome these difficulties a system of scavenging was developed which is now extensively used on large two-stroke-cycle engines. In this system, shown in Fig. 2, ports are used instead of valves. The scavenging ports A are arranged in the cylinder wall near the bottom of the cylinder opposite the exhaust ports B. When the piston reaches the position C in the power stroke, it begins to uncover the exhaust ports B and allow the products of combustion to escape to the exhaust pipe. When the piston has reached the position D, the pressure in the cylinder has dropped nearly to that of the scavenging air and the scavenging ports A begin to open and admit the air to the cylinder. When the piston has reached its final position E, all ports are open wide. The incoming scavenging air is deflected upward by the shape of the piston, so as to drive out all the exhaust gases. On the return stroke, the piston first closes the scavenging ports and then the exhaust ports, after which compression takes place.

This system is slightly less efficient than the valve scavenging system, but it does away with all scavenging valves and greatly simplifies the engine, as only two valves are necessary per cylinder—the fuel-inlet and starting valves—and in some cases only one, in case the starting valve is omitted. With this system the combustion chamber is not of the most

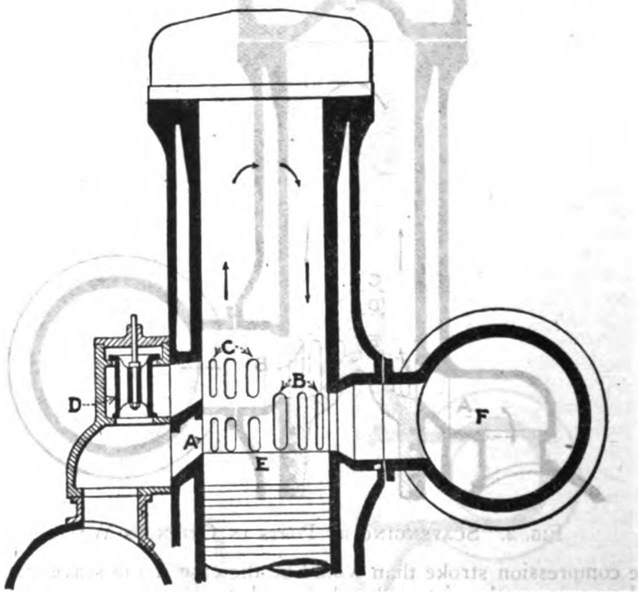


FIG. 3. SCAVENGING BY MAIN AND AUXILIARY PORTS.

efficient form, because of the shape of the piston. It should be as nearly symmetrical as possible and should contain no pockets or cavities. Also the compression is not so high as might be, because the scavenging ports

are closed before the exhaust ports, so that compression starts from atmospheric pressure instead of from 3 to 6 pounds per square inch, as is the case where scavenging valves are used.

Fig. 3 shows the system of scavenging used on an engine of the two-stroke-cycle type. The combustion chamber is not distorted and the scavenging ports are allowed to remain open for a short time after the exhaust ports have been closed. The main scavenging ports A are arranged half-way around the circumference of the cylinder at the bottom, opposite the exhaust ports B. Directly above the main scavenging ports A are the auxiliary scavenging ports C. The admission of air to these ports is controlled by the auxiliary scavenging valve D, which is operated by an eccentric on the engine camshaft. The scavenging ports are inclined so as to deflect the air upward and thoroughly scavenge the cylinder.

As the piston E nears the end of the power stroke it first uncovers the auxiliary scavenging ports C, but no air is admitted through these ports because the valve D remains closed. It then begins to uncover the exhaust ports B, which allows the product of combustion to escape to the exhaust pipe F. When the piston has reached the end of the stroke, it has completely opened the exhaust ports B and also the scavenging ports A, thus allowing air to enter the cylinder from the receiver. At this time the valve D begins to open and admit air through the ports C. On the return stroke the piston first closes the main scavenging ports A, but by this time the valve D is wide open and air is still entering the cylinder through the ports C. Next, the exhaust ports B are closed, shortly after which the auxiliary scavenging ports C are closed. The valve D remains open until the ports C have been covered by the piston.

These three systems of scavenging are the ones generally used on two-

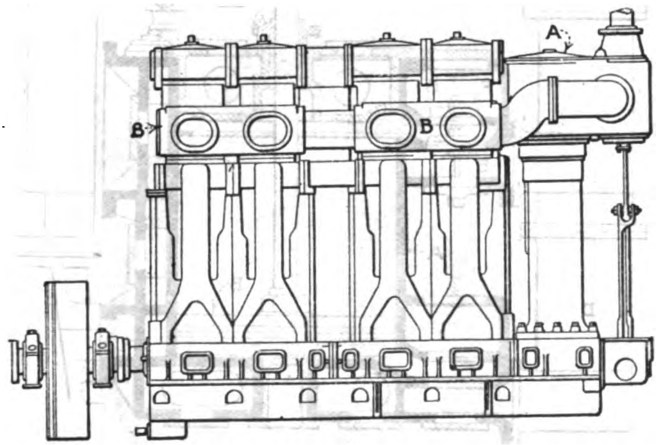


FIG. 4. ARRANGEMENT OF SCAVENGING PUMP AND AIR RECEIVER.

stroke-cycle Diesel engines. There are a few other systems used, but they differ only in minor details from the ones described.

The air for scavenging is obtained from the scavenging pumps, which are of the low-pressure reciprocating type and have a large capacity. The pumps are usually driven by the main engine, either from the crankshaft or by means of levers and rods connected to the cross-heads or connecting-rods. In marine work two pumps per engine are usually furnished,

so that in case one breaks down the other will deliver enough air to keep the engine running, as each pump has about 60 per cent of the total capacity required to run the engine.

The scavenging pumps discharge into a receiver, usually of small dimensions, which runs the whole length of the engine and supplies air to each cylinder for scavenging. This arrangement is shown in Fig. 4, in which A is the scavenging pump discharging into the receiver B which supplies air to the cylinders.

Very often large engines are divided into two sections; that is, a 4-cylinder engine is treated as two 2-cylinder engines and a 6-cylinder engine is treated as two 3-cylinder engines, one scavenging pump supplying each section. On very large double-acting engines one scavenging pump per cylinder is often used.

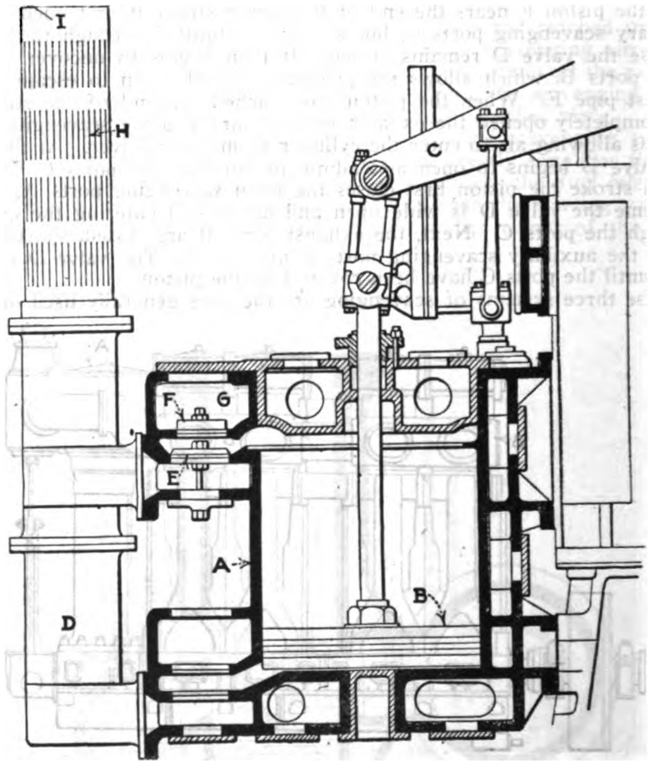


FIG. 5. CROSS-SECTION OF TYPICAL SCAVENGING PUMP.

Fig. 5 shows a cross-section of a typical scavenging pump. The cylinder A is mounted on the engine frame and the piston B is driven by the beam C, which in turn is driven, through links, from the crosshead of the main engine. The air is drawn from the suction pipe D, through the suction valves E, into the cylinder A, and is discharged through the discharge valves F into the passage G which leads to the scavenging air receiver. The flat or grid valves commonly used in this type of pump are closed by

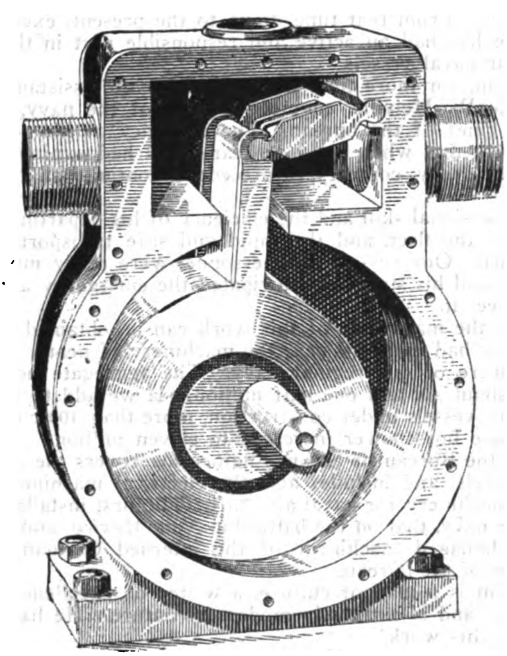
their own weight or by light spiral springs. Air enters the suction pipe D through the vertical slots H in the cylinder I, which acts as a silencer and strainer.

In the designing of scavenging pumps, the most important point is the provision of an ample supply of air, but it is difficult to say what should be the exact capacity of the pumps relative to the working-cylinder volume. That it must be in excess of the latter is generally agreed. For ordinary marine engines and low-speed stationary engines of the two-stroke-cycle type, the volume of the scavenging pump is frequently made 25 per cent greater than that of the working cylinder, while in high-speed engines it is often made 50 per cent greater.

Lately, turbo-blowers have been substituted for scavenging pumps in a number of cases. The results have been satisfactory, as a properly designed turbo-blower is capable of delivering a large volume of air at a pressure high enough for satisfactory scavenging, and it will undoubtedly be used extensively in the future, because of its simplicity and ease of regulation.—"Power."

THOMPSON RETURN-LINE VACUUM PUMP.

In a vacuum heating system the condensation from the radiators, together with the air, is drawn by suction to a pump, where the air is separated from the water, and the hot water is returned to the boiler or to a receiver. Naturally, a vacuum pump should be simple in construction, quiet in operation and durable.



SECTION THROUGH VACUUM PUMP.

The illustration shows a pump that has but three moving members, the crank and the impeller, and a hinged plate, the purpose of which is to separate the suction and the discharge side of the pump cylinder. The surfaces of the impeller and the pump casing are cylindrical, and the impeller is at no time in contact with the casing, a very close clearance being maintained. The crankshaft of the impeller is fitted with ball bearings, and as the crank turns in a clockwise direction, it gives the impeller an eccentric motion. As the water enters the inlet pipe it discharges into the space between the rotor and the casing, and as the rotor revolves this water is carried around until it escapes through the discharge pipe. This pump is being manufactured by the Thompson Manufacturing Co. of Des Moines, Iowa.—"Power."

ADMIRAL GRIFFIN RECEIVES HONORARY DEGREE.

The honorary degree of Doctor of Engineering was conferred upon Rear Admiral Robert S. Griffin, engineer-in-chief of the United States Navy, by the Stevens Institute of Technology of Hoboken, N. J., on June 17. While comparatively few, outside of those intimately connected with naval engineering affairs, are aware of the vast scope of his activities, and especially the responsibilities carried during the period of the war, it will be a source of gratification to his friends and associates to learn of this richly deserved honor.

The new navy may be considered as commencing with the Roach cruiser about 1885, at which time Admiral Griffin was an inspector of machinery for those vessels. From that time down to the present, except for tours of sea duty, he has had an active and responsible part in the equipment efficiency of our naval vessels.

Admiral Griffin, for more than five years, was the assistant of the late Admiral George W. Melville, engineer-in-chief of the navy, and in 1913 was appointed chief of the bureau. The efficiency of his administration has been so great that when the four-year detail had expired in 1917 the Secretary of the Navy, contrary to precedent, continued him as chief of the bureau.

Upon his professional skill and the efficiency of his department depended the readiness of the fleet, and the rapid and safe transportation of our armies to France. Our vessels and personnel were ready mainly because Admiral Griffin and his staff had anticipated the emergency and had plans completed to meet it.

Some idea of the magnitude of this work can be obtained by considering that the navy had under its care the machinery of nearly 2,000 vessels, of which 570 were of our regular navy. The aggregate horsepower of these ships is about six and one-half million. If we add to this the total of the number of vessels under construction, more than 400 could be added and the aggregate horsepower increased to eleven million.

The work of the Bureau of Steam Engineering covers the motive power of all naval vessels, and includes also the electrical machinery, the radio telegraph and machinery for naval air craft. The first installation of electric drive in the navy, that on the battleship *New Mexico,* and the rehabilitation of the damaged machinery of the interned German ships, came within the scope of the bureau.

Admiral Griffin is a man of culture, a writer of excellence upon engineering subjects, and is extremely modest by nature. He has never been one to advertise his work.

He was born in Richmond, Va., in 1857, and graduated from Annapolis in the class of 1878.—"Power."

FUEL OIL.

Mr. A. F. Baillie on its Production and Applications.

Mr. A. F. Baillie, of the Engineering Department of the Anglo-Mexican Petroleum Company, Limited, read a paper on "Fuel Oil: Its Production and Applications," on February 8, before the Birmingham Association of Mechanical Engineers.

After dealing with the origin, history and chemistry of petroleum, he pointed out that fuel oil was frequently spoken of as "crude oil," but this is incorrect, as the crude oil—that is, the petroleum oil as it issues from the well—contains certain light fractions of the gasoline and kerosene series, which it is not only desirable to extract from the crude by distillation on account of their higher values, but which if left in the fuel oil and used as fuel, would be a wasteful procedure, and, further, would lower the flash-point, which has been very properly fixed by British authorities, viz.:—Lloyd's Register of Shipping, 150 degrees Fahr.; British Admiralty, 175 degrees Fahr.; London County Council, 175 degrees Fahr. The crude is therefore subjected to what is known as a "topping process," which is a distillation which cuts out those lighter fractions, leaving a product having a calorific value of about 19,000 B.T.U.'s per pound.

The physical and chemical tests of Mexican fuel oil are as follows:

Specific gravity	under .95 at 60 degrees Fahr.
Flash point (open)..........	over 150 degrees Fahr.
Viscosity....................	under 1,500 seconds (Redwood) No. 1 at 100 degrees Fahr.
Calorific value...............	over 18,750 B.T.U.'s.
Sulphur percentage..........	approximately 3.5 per cent.

PIPELINES, STORAGE AND REFINING.

The production of oil is only the first link in the long chain of operations. The crude oil has to be transported to storage tanks at the refineries, which are generally situated on seaboard to facilitate shipment of the refined products. For this purpose steel pipelines are employed of 8 or 10 inches diameter, with powerful pumps at intervals along the line, and in this way crude oil is delivered to refineries from wells which are many miles away. One of the Mexican Eagle Oil Company's refineries, situated at Minatitlan, is capable of turning over 1,000 tons of crude petroleum daily into a complete range of high-grade products, *i.e.*, gasoline, kerosene, lubricants, fuel oil, paraffin wax, and asphalt for roads. Another refinery has recently been completed at Tampico, and has a total throughput capacity of 3,000 tons of petroleum per day.

LOADING, BUNKERING AND MARINE TRANSPORTATION.

Fuel oil is loaded and ships are bunkered in Mexico at Tampico, Tuxpam and Coazacoaleos. At Tampico and Coazacoaleos the operation of loading is by means of a series of 8-inch pipelines leading from the refinery pumping station to the steamer's jetty on the river. At Tuxpam, however, owing to the fact that the water inside the bar is too shallow for the large 15,000-ton Eagle Oil Transport Company's steamers to come alongside and load, it was found necessary to provide sea loading-lines. Accordingly, submarine pipelines have been laid on the bed of the sea, reaching out to a loading terminal about 1½ miles from the shore. The pipelines are then connected by flexible hose to the steamers lying at these moorings, and three or four vessels can be loaded at once from the storage tanks and pumping station on shore. On an average each steamer is

loaded and despatched within two and a half days, as the pumping and sea-line facilities enable a ship to be loaded at the rate of 10,000 tons per 24 hours.

The next step is the transport from the refineries to the world's markets, for which purpose a large fleet of tank steamers, of over 15,000 tons deadweight capacity, have been built by the Eagle Oil Transport Company. The latest addition to this fleet is the *San Florentino,* which has a deadweight carrying capacity of 18,000 tons, and is the largest oil tank steamer in the world. This vessel, which has a greater displacement than a dreadnought, was recently launched on the Tyne, and is now being fitted with geared turbines and oil-fired boilers.

The tanks of these vessels are fitted at the bottom with heating coils, which enable them to transport not only ordinary grades of fuel oil, but also the heaviest Mexican fuel oils, in bulk during the winter months, the fuel oil being loaded in Mexico at a temperature of 100 degrees Fahr., allowed to cool down to about 85 degrees Fahr. during the voyage, and when the vessel is about three days from the home port these steam heating coils are brought into play and the temperature again raised to about 100 degrees Fahr., to facilitate speed of discharge. Four large steam-driven cargo pumps are installed, and are fitted as close to the bottom of the ship as possible, so as to obtain the minimum suction lift. These pumps are capable of handling 1,200 tons, or 7,500 barrels of oil per hour.

The tank steamers discharge their cargoes into ocean storage installations at various ports in this country and abroad, into storage tanks ranging in capacity from 30 to 8,000 tons. The largest tanks in this country have a capacity of 6,600 tons, and are 95 feet in diameter by 37 feet high.

APPLICATIONS AND ADVANTAGES.

From a rapid survey of production and transport he passed to a summary of the applications of fuel oil and its proved advantages in comparison with coal.

(*a*) *Warships.*—During the war many thousands of tons of oil fuel have been supplied to the Admiralty. Not only has it been used in submarines, torpedo destroyers and fast cruisers, but the newest type of capital ships, such as the *Queen Elizabeth* series, are fitted to burn oil fuel only. Its use has without doubt contributed to our naval supremacy, notably in the battle of Falkland Islands. In 1915 Lord Fisher, the veteran advocate of an oil-fired navy, wrote as follows: "Machinery has no nerves, and the all-oil boilers do not get tired. No more hell in coaling ship. The unfailing propellers get us at top speed on the spot to a minute, though 7,000 miles away." This reference is obviously to the Falkland Islands battle, which was won in December, 1914.

The points in favor of oil in place of coal for warships are:—

(1) Radius of action increased by 50 per cent on equal bunker weight, or 80 per cent on equal bunker space.

(2) Up to 83 per cent thermal efficiency, instead of 60 per cent.

(3) Boilers can be forced up to 50 per cent above normal rating.

(4) Control of smoke: entire absence or dense smoke screen as desired.

(5) Reduction of labor by about 70 per cent.

(6) Constructional advantages.

(7) Bunkering at sea.

(*b*) *Mercantile marine.*—There are many equally attractive advantages in the application of oil to the mercantile marine. One interesting comparison on coal and oil was made by the American Hawaiian Steamship

Company's SS. *Arizonan*. This vessel took 161 days on the round voyage when burning oil, as compared with 186 days when burning coal, and by economies effected and increased earnings due to saving of bunker spaces, and decrease in operating staff, she saved $19,500 on the voyage, or over £4,000.

It is important to remember that the furnace of a vessel burning oil may be readily altered to burn coal, or *vice versa,* when required. If suitable arrangements are made the work can be done at the cost of a few pounds by the ship's own staff without any considerable trouble or delay to the steamer. The advantage offered by this ready conversion is that with vessels which trade between countries where either coal is dear and oil cheap, or the reverse, the cheaper fuel can be used on both journeys, resulting in remarkable economies. The Toyo Kisen Kaisha steamship line is operated in this way.

Another instance is a comparison of the Eagle Oil Transport Company's vessels *San Dunstano* and *San Eduardo,* both vessels of 9,000 tons deadweight carrying capacity, which are fitted to burn coal and oil respectively. The consumption of fuel worked out at two to three in favor of oil, while the I.H.P. developed showed an 18 per cent improvement in the case of the oil-fired vessel. The striking fact of the comparison is that the *San Eduardo* made the round voyage to Mexico and back eight days quicker than the other, making a large improvement in her earning capacity.

On a comparative test run at Wallsend Slipway on coal and oil on a Scotch marine multitubular-type boiler the following results were obtained:—

MEXICAN FUEL OIL.

Fuel Oil—	
Specific gravity at 60° F.	.953
Viscosity at 100° F. (Red. No. 1)	2,130 secs.
Flash point (close)	160° F. (above)
Calorific value	18,430
Water evaporated—	
Lbs. of water per lb. of oil	12.15
Lbs. of water per lb. of oil (from and at 212° F.)	14.38
Boiler efficiency	73.37 per cent

COAL.

Coal—	
Calorific value in B.T.U.	14,432
Water evaporated—	
Lbs. of water per lb. of coal	7.76
Lbs. of water per lb. of coal (from and at 212° F.)	9.31
Boiler efficiency	62.28 per cent

Efficiencies as high as 84.5 per cent have been obtained in this country on Scotch marine type boilers, using Mexican fuel oil with the pressure system of oil burning.

At a large London factory which converted their Lancashire boilers from coal to fuel oil firing, the water evaporation per pound of coal having a calorific value of 11,451 B.T.U. was 7.22 pounds, whereas when working with the pressure system of oil burning, using an oil having a calorific value of 18,750, the evaporation per pound of oil reached 14.44 pounds. The quantity of water evaporated per square foot of heating surface on coal was 3.3 pounds, whereas oil showed over 7 pounds, thereby increasing the boiler rating by over 100 per cent.

METALLURGICAL FURNACES AND BURNERS.

Fuel oil in this country is mainly used in industrial furnaces, such as billet heating, metal melting, riveting, bolt and nut making, and the manufacture of glass bottles.

The largest quantity of oil fuel used in 1917 for furnace purposes was for billet heating in connection with the manufacture of shells. In pre-war times the principal consumers were glass-bottle manufacturers and rivet, bolt and nut makers. In post-war times we may see a drop in the consumption of billet heating, but we hope to see a large increase in the consumption for glass manufacture. Again, large users of oil fuel are found in the metal-melting industry, melting metal from the low-fusion aluminum to the high-fusion steel. The melting-points of these range from 1,000 degrees to 3,000 degrees Fahr. (approximately 540 degrees to 1,650 degrees Cent.) In the manufacture of wrought iron materials these are usually forged to drop stamped, and for this purpose it is necessary to heat the metal up to a temperature of approximately 2,500 degrees Fahr. (approximately 1,370 degrees Cent.). This heat was formerly obtained by means of placing iron bars in a coke furnace of the type used by the wayside smith, but of larger dimensions and with improvements. This, until gas or oil fuel was introduced, was rather a slow process, and since oil fuel has been used in an intelligent manner the output of each new or converted furnace has been almost trebled.

Another point which is worthy of notice and which usually appeals to every works manager, is the fact that when utilizing oil fuel the size of the furnace is reduced by 30 per cent, and the working area occupied by a furnace and its adjacent machines is only about 50 per cent of that of the coal-fired furnace, as a stack of coal occupies a certain amount of space in front of the furnace, and the resultant ashes occupy another space behind the furnace.

The next interesting manufacture is that of metal melting. The principal metals which have been melted during the war are aluminum and what is known as "70/30 brass," aluminum being principally used for aeroplanes and motor car work, 70/30 brass being principally used for cartridges.

AUXILIARY TO COAL AND GAS.

In the early years of this century a French engineer introduced the subject of oil as an auxiliary fuel, a combination to which little attention had been paid in the past. The main advantages of auxiliary firing, he remarked, lay in being able to obtain at will a large increase in the power of boilers, also that the combustion of the petroleum does not in any way prejudicially affect that of coal. In fact, by the introduction of jets of petroleum the condition and efficiency of combustion are improved by more completely mixing the gases. It is therefore not correct to consider the evaporative power of coal as identical, when passing from ordinary to auxiliary firing. Admitting this as a principle, and supposing the quantity of water evaporated by the coal to be constant, the extra evaporation, due to the better mixing of gases, is credited to the petroleum.

Several evaporative trials were made with coal only and coal and oil firing on the same boiler of a French Navy ship, with the following results:—

Test No.	Fuel Used Coal Per cent	Oil Per cent	Lbs. of fuel per sq. ft. of grate area	Water evaporated per lb. of fuel used	Remarks
1	100	..	18.8	9.05	
2	55	45	21.3	11.34	25 per cent evaporative increase over No. 1.
3	36	64	21.7	14.12	56 per cent evaporative increase over No. 1.

These tests were encouraging, and should have served as a basis for further experiment, but the question was not developed until about four years ago, when power station engineers started taking a keen interest in the subject. Their theory was that a poorer class of coal could be used in conjunction with oil fuel than could be burned satisfactorily by itself under the boilers, this being due to the fact that the poor class of coal tended to cake on the links of the chain grate stokers, thereby retarding the necessary quantity of air being drawn through the bars to complete combustion.

The result was that the poor classes of coal were merely covered with a smoldering mass, which traveled slowly along the bars and were dumped into the ashpit. This ash, on analysis, contained a very high percentage of partly consumed coal, thereby causing a very much higher quantity of coal to be burned to maintain any rated evaporation that would have been used if a better-class coal were employed.

When auxiliary fuel oil is applied, the theory is that as a result of the almost perfect combustion obtained thereby, the combustible gases rising from the coal fuel bed were quickly ignited causing the top of this mass to become much more incandescent, thereby tending to aerate the bottom mass, which would then allow sufficient air to be drawn through to complete the combustion of the rest of the coal.

Sufficient interest was taken in this theory for a large London power station to give their sanction for tests to be carried out under one of their coal-fired Stirling water-tube boilers. One burner was introduced into each side of the boiler, approximately about 25 per cent from the back of the grate, the burners being opposite one another. The fuel oil was stored in an overhead tank, capable of holding three or four days' supply. The oil then gravitated to the burners, which were of the Kermode steam jet type. operating with steam as an atomizing agent at a pressure of about 25 pounds per square inch.

The final of a series of experimental mixed-fuel tests was carried out on a nutty slack, having a calorific value of 10,300 B.T.U.'s and Mexican fuel oil, having a calorific value of 18,750 B.T.U.'s. A boiler efficiency of 74 per cent was obtained, and the temperature of the combustion chamber was 2,850 degrees Fahr. and uptake 628 degrees Fahr. The proportion of oil to coal on a B.T.U. basis was 8 per cent and on a weight basis 4.96 per cent.

Assuming coal at 20s. 10d. per ton and oil at 50s. per ton, the cost per ton of water evaporated using coal only was 2.8s., and using the aforementioned percentage of coal and fuel oil 2.62s., showing a monetary saving of 6 per cent.—"Shipbuilding and Shipping Record."

THE EXAMINATION OF MATERIALS BY X-RAYS.

The general discussion on "The Examination of Materials by X-Rays" by a joint meeting of the Faraday Society and the Röntgen Society, held in the rooms of the Royal Society recently, clearly brought out the great possibilities as well as the limitations and difficulties of this promising

method of testing materials, which excites far wider interest than might be supposed from the sporadic notes and papers so far published. Great credit is due to Sir Robert Hadfield, Bart., F.R.S., for the initiation of this discussion over which he presided, as president of the Faraday Society, supported by Dr. Batten, the president of the Röntgen Society. The meeting was opened at 5 p.m., resumed after a dinner interval at 8:30, and did not disperse till near 11 o'clock. That is nothing unusual for the Faraday Society. As it was, the remarkably good collection of exhibits by scientists and firms did not receive the attention it deserved, because the instructive character of the discussion kept members within the lecture hall, though several speakers were restricted by official ties. With a very natural pride, Sir Robert Hadfield dwelt upon the fact that this was the twentieth general discussion arranged by the Faraday Society, nearly all as successful and crowded as on last Tuesday, when the Society for the first time enjoyed the hospitality of the Royal Society. This is a remarkable record for a comparatively young society of a little more than 300 members. As Sir Robert has occupied the chair during the whole war period, he has presided over a good many of these meetings to the success of which, it should be added, he has contributed in manifold ways. Mr. F. S. Spiers, B.Sc., to whose valuable services Sir Robert expressed the society's indebtedness, has been secretary since the foundation of the society, the offices of which are at 82 Victoria street, Westminster.

In his introductory remarks Sir Robert Hadfield said that, when suggesting a general discussion on Radiometallography, he had been quite aware that such an examination for flaws might, if unfairly used, cause manufacturers great trouble and expense, and he had certainly not wished to cast any doubt on the wonderful improvements of the last decades in metallurgical production. His own firm, he might state, had been turning out 8,000 9.2-inch high explosive shells weekly at one time during the war—other firms had done similar big things—and in spite of all the pressure and hurry there had not been a single instance of their products failing through defects; that applied also to some 6,000 guns and tubes for guns, etc. But technologists had to study every new means of ensuring perfect materials. So far X-ray examination seemed limited to thin sections of metal, though some experimenters had dealt, in this country, with specimens of 4-inch and even of 9-inch thickness. Sir Robert had himself submitted to Professor Bragg a large number of purest specimens obtainable in the hope that the X-ray study would clear up the differences in the physical properties of the metals of the iron groups, and the magnetic and other peculiarities of critical points and of allotropic modifications. Those specimens included pure electrolytic iron, various kinds of steel, overheated, cast, etc., several alloys, and manganese metal, cobalt, chromium, tungsten, molybdenum in various conditions. Proceeding to a review of the literature on the subject, which he had conveniently summarized in a bibliography at the end of his paper, Sir Robert drew attention to the X-ray examination of flaky steel by H. S. Rawdon, of Washington—we had a Note on this subject in our issue of last week—to the equipment for X-ray examination which the firm of Henri Pilon had supplied to Messrs. Schneider and other firms; and to two recent German publications on the subject, which Sir Robert had had translated on account of their generally instructive and practical character. Professor Porter subsequently gave an abstract of these papers, copies of which were distributed. Sir Robert Hadfield's own contributions to the researches will presently be mentioned.

In his opening address, Professor W. H. Bragg, C.B.E., F.R.S., of University College, London, stated that it was one of the chief purposes of the conference to display the possibilities of X-ray methods to those who

had no occasion to make a study of X-rays themselves. X-rays were of the same nature as light, but of extremely short wave-lengths. Whilst ordinary light gave us information relating to the outward form, the X-rays passed right through a body and revealed its interior condition. For practical purposes X-rays were excited by causing a powerful electric discharge to take place within an exhausted bulb (Coolidge tube), in which the discharge fell on a block of metal, the target or anti-cathode. The actual impact was limited to a small area of the target, the focus; a large amount of the energy unfortunately went into heat, and ingenious devices were adopted to get rid of that heat. The quality of the rays depended upon the target material, tungsten or platinum in the Coolidge tube, but not upon its temperature, and though even rays from tungsten were not quite satisfactory in quality, we had to use such refractory metals which would stand the rise of temperature. For good definition of the radiographs, the focus should be "fine"; but a fine focus would not take much power. The rays emitted from the target in all directions had three main effects; they excited photographic plates and phosphorescent screens, and they ionized gases. The last effect was useful in the laboratory, the second in surgery, but hardly in metallurgy and engineering, and the action on photographic plates would probably be the basis of engineering practice.

The X-rays had the extraordinary property of passing through all substances more or less. The shortest, hardest rays, produced by the highest potentials, had the greatest penetrating power, and might alone prove useful in engineering. The transparency of a material to X-rays decreased as the atomic weight increased. Heat had no influence; a red hot casting gave the same radiograph as a cold casting. But owing to that high penetrating power the rays could neither be focussed nor deflected by lenses or prisms. The penetration depended merely on the atomic weight and the thickness of the material. Professor Bragg exemplified this by a selection of beautiful radiographs. The positive of a watch clearly showed all the parts, wheels, springs, etc.; in a cherry blossom the delicate nerves of the petals were quite distinct; a tadpole which had swallowed a worm looked less dark than the worm which somehow seemed to consist of substances of higher atomic weight; in an electric kettle heater the wire, the porcelain and the star-shaped iron support all cast their shadows. In some photographs of good and faulty castings and welds the tool marks were quite distinct; considering how minute the differences in thickness due to the tool marks were, that fact indicated the truly remarkable sensitiveness of the method.

The explanation of that sensitiveness depended mainly on two considerations. When two X-ray beams of equal initial intensity passed through two slabs of the same material of different thicknesses, the ratio of the transmitted intensities depended only upon the absorption in the extra thickness of the one slab, but not at all upon the absolute thickness. If we imagined two beams to strike the photographic plate, the one passing first through steel, the other striking the plate directly, the intensity ratio of the transmitted beams, $T_1 : T_2$, was dependent upon the absorbing power of the steel; if further two equal blocks of some material were interposed in the two beams, both T_1 and T_2 would be reduced, but their ratio would remain the same. Secondly, we had to consider the way in which the photographic image was formed. The blackening of the plate, due to two rays of unequal intensities, increased with the intensity of the rays. The transparency of the plate diminished as the blackening increased, but the ratio of the transparency in two spots, the "contrast" on the plate, was not the ratio of the two ray-intensities. Supposing we had two rays, one twice as intense as the other, and that in the case of the

stronger ray, the passage for each period of time reduced the transparency to half its value. In the case of the weaker ray it would always take two periods of time to reduce the transparency to half its value, and when the two transparencies were compared at any time they would be found not to be always in the ratio 2:1. By under-exposure and over-exposure any desired contrast might be obtained, but other circumstances came in, and the plates might become too black; yet by proper manipulation visible contrasts would be secured. Professor Bragg mentioned that Dr. Slade would deal with these points more fully.

Professor Bragg then returned to the question: should hard or soft X-rays be used? For penetrating through the material the rays should be hard; but then they penetrated also through the plate, and we had to increase their action on the plate. That action was due to the liberation of electrons which darted to and fro among the surrounding atoms until their energy was spent. In the emulsion, electrons were set in motion mainly through absorption of X-rays by the silver and bromine, and that action might be reinforced by putting heavy atoms into the emulsion. Some experimenters placed a thin lead sheet on the film, so that the electrons generated in the lead struck back into the photographic plate. The variation in the absorption of rays of different wave-lengths, studied by the Duc de Broglie and others, suggested a further consideration. As wave-length increased the absorption by a given substance also increased; but at a certain value, critical for each substance, the absorption suddenly dropped, to rise again steadily afterwards. There was, therefore, a "step down" in the photograph—it should not be called a band—marking the critical wave-length for silver. A similar "step up," on the side of short wave-length, would be obtained by interposing an absorbing screen, say of antimony or some other metal, in the path of the rays. That "antimony line" was so sharp that it might serve for a new method of spectroscopic examination of the constituents of a material. We might also put some special material in the emulsion and choose X-rays intermediate between the actual value of this material and of the casting under test; but this did not appear practicable. Professor Bragg finally referred to the examination of crystal structures by X-rays, on which he has himself done pioneer work, and which has practical interest to metallurgists. The war, he concluded, had interrupted much X-ray research which might soon be resumed, he hoped.

A paper sent by Mr. E. Schneider, on "The Industrial X-Ray Examination of Metals at the Laboratories of Messrs. Schneider, Le Creusot," was then read in abstract. The radiographs, taken with the apparatus and Coolidge tubes of Messrs. Pilon, exemplified the remarkable influence of aluminum in eliminating blow-holes in fish-plates; the examination of steels containing different percentages of tungsten; and the examination of compound metals. A fault in a cast steel bracket of a gun carriage was revealed by X-rays and the method changed to avoid repetition of such flaws. The steel specimens so far tested do not exceed 45 mm. in thickness; but that limiting thickness is reduced when the steel contains a constituent of higher atomic weight.

Of the two German papers, already referred to, which Sir Robert Hadfield had had translated and which Professor A. W. Porter, F.R.S., of University College, London, abstracted, the one by Dr. Respondek, of Halensee, near Berlin, outlined "The Principles Governing the Penetration of Metals by X-Rays"; and the other, by F. Janus, of München, and M. Reppchen, of Köln, explained the "Investigation of Metals by Means of X-Rays." The latter paper was based upon researches conducted in the laboratories of the firm of Reiniger, Gibbert and Schall, A.G., of Berlin-Erlangen, and was in particular compiled for the technical man

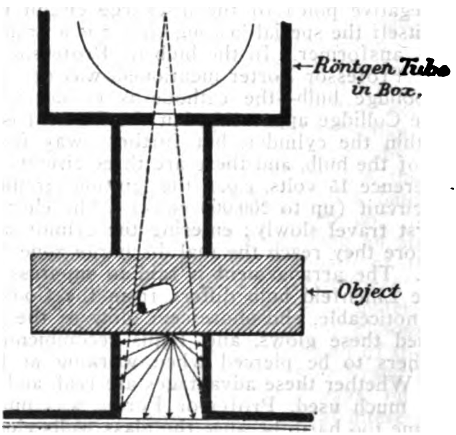
unfamiliar with Röntgen-ray bulbs. Perhaps we had better adopt a general tone in our notice, since Professor Bragg had purposely left several points to subsequent speakers.

In the ordinary Röntgen bulb (gas tube) positive ions striking the cathode drive out electrons from the cathode, which, in their turn, hitting the anti-cathode or target, give rise to the emission of X-rays from the target. For the examination of materials we want hard, uniform rays. When we depend upon the ionization of the gas molecules in the bulb, however, there will be soft as well as hard rays, and the hardness will change during the working. Hence bulbs of the Coolidge type came into use; in these the vacuum is pushed as high as possible. There are hence few gas molecules left that might be ionized, and the stream of electrons required for the causation of the X-rays is obtained by heating to incandescence a Wehnelt cathode or a thin wire of tungsten (the spiral or filament) by a special current; this spiral is surrounded by the cathode proper (the negative pole) of the discharge circuit in the bulb. In the Coolidge tube itself the special heating circuit is a branch of the secondary circuit of the transformer. In the bulb of Professor J. E. Lilienfeld, of Leipzig—which Professor Porter mentioned, was described in 1912, a year before the Coolidge bulb—the cathode is cylindrical and opposite the target as in the Collidge apparatus; but the filament is of large size; it is not placed within the cylinder, but further away from the target, in a lateral branch of the bulb, and there are three circuits—the heating circuit (potential difference 15 volts, *e.g.*), the ignition circuit (2,000 volts), and the discharge circuit (up to 200,000 volts). The electrons emitted by the hot cathode first travel slowly; entering the cylindrical cathode, they are accelerated before they reach the real discharge zone between the cathode and the target. The arrangement is said to suppress soft rays from the outset, and the Lilienfeld bulb differs from the Coolidge bulb as to the peculiar, little noticeable, phosphorescent glow of the glass walls. Lilienfeld has studied these glows, and Janus recommends his bulbs as less likely than others to be pierced when working at high potentials and temperatures. Whether these advantages are real, and whether the Lilienfeld tubes are much used, Professor Porter was unable to say. When gas bulbs become too hard, because the glass walls electrostatically attract the gas (air) molecules and remove them from the bulb, some air is admitted into the bulb from a pocket. This can be done automatically by the aid of a milliamperemeter acting as a relay. In any case, the discharge and hardness are regulated with the aid of an electrometer (called also penetrometer in this connection) and a milliamperemeter.

Light radiation passing through a substance will partly be reflected; part is absorbed, part scattered and part transmitted. The absorption of X-rays increases proportionally to the atomic weight in accordance with $J = J_0 e^{-\lambda d}$, where J is the intensity of the absorbed radiation, J_0 that of the incident radiation, e is the logarithmic base, λ the coefficient of absorption, and d the thickness of the substance. Instead of λ, μ is sometimes used, being defined by $\mu = \lambda/\rho$, where ρ is the density; the formula is then $J = J_0 e^{-\mu\rho d}$. The absorptions may also be expressed in half-values (similar to the half-life period of radioactive substances), the half value indicating the thickness of the respective material which would stop half the radiation, expressed in centimeters (or fractions of a centimeter) of a foil of aluminum of equal stopping power. The following table summarizes the atomic weights A.W., densities ρ, μ/ρ and half values of a few metals for hard rays.

In order to secure sharply-defined radiographs the rays falling upon the specimen should be parallel and normal to the plate. As a matter of fact, the bundle of rays is conical, and the scattering still more impairs the

definition. The scattering takes place in all directions within the specimen and is not confined to the specimen, of course, but the whole air space, and objects in the room help to scatter; rays also fall upon the plate and table by the side of the specimen, and enter the plate from the sides and from below. In order to reduce the scattering the bundle of rays is confined within a tube of lead, and sheets of lead are applied to all the parts of the plate which might be excited by stray radiations. The diagram annexed (from the paper by Janus and Reppchen) illustrates the action in the case of thick specimens. The rays pass through a diaphragm in the bulb casing of lead, down the upper lead tube, through the specimen, down the lower tube and enter the plate, first passing through the intensifying screen which is interposed between the plate and the top of the plate carrier.



	A-W.	ρ.	μ/ρ	Half-Value.
Aluminium ..	27·1	2·6.	0·97	0·28
Iron	55·8	7·7	7.5	0·012
Silver	107·9	10·5	17.7	0·0037
Platinum ..	195·2	21·5	22	0·0015
Mercury	200·6	14·4	22	0·0022
Gold	207·2	11·4	22	0·0028

The secondary radiation from below is particularly troublesome with thick specimens; if lead sheets, up to 10 mm. thick, cannot conveniently be used, cast lead may be employed, or lead-shot be piled on the parts. For the protection of the operator the bulb is placed within a brick chamber which is fitted with a window of one or several layers of lead glass; switches are mounted on the outside of the wall; further information on these points was given in the paper by M. Pichon.

The photographic films must be thick and rich in silver; a slow developer (glycocoll) has to be used, as a rapid developer would fog the surface. The exposure may have to be long, but can be much shortened by the aid of intensifier screens (*e.g.*, of calcium tungstate), which are placed

over the plate. The blue light of the tungstate reduces the exposure, to one-sixth and even much smaller fractions; these screens are delicate, however, and easily stained; they impress the grain of the tungstate crystals on the radiographs and are, according to Janus, apt to impair the definition; the point was not raised during the discussion. With a distance of about 30 mm. between cathode and target, the specimen should be about 40 cm. below the focus of the bulb. In the examination of tubular specimens, the films are placed inside the specimen. Blowholes come out much better than cracks; unless parallel to the rays cracks are hardly shown by the X-rays. Faults near the upper surface appear more distinct than faults deeper down. The position of a flaw can be fixed in various ways. A simple method, not always applicable, however, is to take two radiographs on the same plate, shifting the plate laterally for the second exposure. During the exposure the bulb and apparatus must, of course, be kept perfectly steady, a condition not easily realized in works. In ferro-concrete any deformation or rusting of the iron bars can easily be detected. The clean iron gives sharp shadows, the rusty iron thicker, more hazy shadows; Respondek gave a good photograph of a ferro-concrete slab, 20 mm. in thickness, exposed to the rays for five minutes. X-ray testing of the quality of a cement, and the study of the chemical changes going on in cements, have so far failed.

A series of communications on researches carried out by Sir Robert Hadfield, Mr. S. F. Main, B.Sc., and Mr. J. Brooksbank, B.Sc., in the Hadfield Research Laboratory, Sheffield, were then presented. We hope to reproduce some of the photographs next week. The first on "Testing the Absorption Power of Different Steels Under X-Rays," dealt with some of the early experiments. Discs, 1 inch in diameter, one-thirty-second of an inch thick, were placed either on a barium-platinocyanide screen or on a photographic plate. The specimens were a very pure iron (0.03 C, 0.01 Si), density 7.78, forged; a steel with 1.19 C, 0.07 Si and 2.45 Al, density 7.57; and an alloy steel (0.55 C, 2.79 Cr, 20.4 W), density 8.77; the permeability decreased as the density increased, but the differences were not sufficiently marked when the fluorescent screen was used, and the addition of an intensifier proved advisable in the case of the plate. Similar discs, all of the very pure iron, were then prepared, increasing in thickness in steps of 1/64 inch from 1/32 inch up to 3/16 inch; a piece of thin, irregularly bent iron wire was placed over each disc to help bringing out contrasts; some other steels were tested at the same time. The thicker discs gave darker pictures (all negatives); but a hardened quenched steel disc produced a less dark negative than the annealed specimen. This was contrary to expectation.

The second paper on "X-Ray Examination as Applied to the Metallurgy of Steel," was of a general kind. Steel specimens up to 4 inches in thickness, it was pointed out, were said to have been successfully examined. To be of actual service the method should allow of examining articles of practical sizes; otherwise the chief advantage of the method, its non-destructive character, would be lost. The examination might be applied to routine examination of castings and forgings in the course of their manufacture, and also to the materials used. Defects such as pipe reproduce themselves under similar conditions of manufacture; but many defects in steel were of a casual nature. Such routine work would much be facilitated if the necessity for photographing could be avoided, and fluorescent screens could be used; for inspection work in particular a simple screen method would be preferable. At present this did not appear to be possible, and the appliances were too costly and delicate. A point to be borne in mind was that the X-ray examination did not magnify the defects to be looked for, which had to be visible to the naked eye. On

the other hand, the method gave us information about the atomic and crystalline structure, and might be useful in the investigation of allotropy as also in metallurgical analysis.

The third paper, on "The Radiographic Examination of Carbon Electrodes Used in Electric Steel-making Furnaces," referred to the troubles caused by the fracture of the electrodes, which may spoil a whole cast of steel. The carbons, up to 20 inches diameter, are made of amorphous carbon or of graphite; the former, which gave most trouble, were more heterogeneous than graphite and more suitable for radiography, it was stated, and the paper dealt particularly with amorphous carbon electrodes. In the electrode manufacture, retort carbon, anthracite or petroleum coke or mixtures of them are crushed and mixed with pitch and tar to be moulded under pressure, in cylinders or by extrusion, and then baked. The paper contains particulars as to diameter, ash contents (generally about 4 per cent, 15.8 per cent in one case), actual and apparent gravity (ranging from 1.8 to 2.2 and 1.47 to 1.65), and porosity (3.5 per cent to 33.8 per cent, the carbon mentioned rich in ashes having a porosity of 11 per cent), of eight samples from different firms. Sections, 1 inch in thickness, were examined by Messrs. H. W. Cox and Co., with a Coolidge tube, 6-inch spark gap, 20 inches distance anode to plate, 2 milliamperes and exposures of 1 minute, which might have been reduced. Positives were afterwards prepared from the negatives in the Hecla Works of Messrs. Hadfield, and these positives are much clearer, it was stated, than the negatives reproduced in the paper. In one set of experiments small boxes of the materials used, retort carbon, pitch, and anthracite were radiographed together with the respective electrodes; those pictures afforded little further information, probably because the baking altered the raw materials too much. The information gained as to the structure was considered valuable, though the interpretation of the radiographs did not appear to be easy. A coarse structure was not found detrimental to good behavior, *i.e.*, freedom from fracture; the specimen richest in ashes [a point to which attention is not drawn in the paper, we should say] proved the best electrode; it was of a very coarse structure, but the worst electrode was also very coarse, and the electrode of finest grain proved only of average quality. A systematic investigation, the authors stated, of electrodes of various compositions, baked at different temperatures, etc., appeared desirable and might explain the appearance in the radiographs of some very dark grains, possibly grains in the process of assimilation. Some graphite electrodes gave radiographs devoid of any structure; in one of them a crack was shown.

The fourth paper from the Hadfield Research Laboratory was on "A Method of Testing an X-Ray Tube for Definition." We shall later notice this paper together with others of a similar nature.

In a short note on "The Detection of Hair Cracks in Steel by X-Rays," Lieutenant-Colonel C. F. Jenkin stated that he had been persuaded to try whether it would be possible to detect hair cracks which had given considerable trouble in the crank-shafts of aero engines, though there seemed to be little likelihood of success. The samples had been examined by the experts who were pushing the process. The samples were cut from crank-shafts with large and relatively obvious flaws; but no trace of defects was shown by the X-rays. One sample, about 2 cm. thick, was cracked right through the thickness for a length of 6 cm.; but the radiograph gave no indication of the crack. If such large flaws were undetectable by the X-rays, the method was obviously useless for finding hair cracks, Colonel Jenkin remarked, adding that he did not follow Professor Bragg's arguments concerning the explanation of the detection of minute differences in the sheet thickness by means of the rays.

RADIOMETALLOGRAPHY.

The appliances used at Creusot are supplied by Messrs. Pilon, and further information on these apparatus and their use was given in a paper on "Radiometallography" by Mr. Hector Pilon and Mr. Geoffrey Pearce, of Messrs. Watson and Sons, Sunic House, Kingsway. The paper was read by Mr. Pearce, who stated that they had been able actually to penetrate steel in thicknesses up to 55 millimeters; with greater thickness the exposures became too long.

Under similar conditions, the same energy input, same potential and current, same distance, etc., a single block of a metal requiring an exposure of 30 seconds for a thickness of 15 millimeters, would require 250, 2,000, 3,500, 4,800, 6,500 seconds for thicknesses of 25, 40, 45, 50, 55 millimeters; thus a few millimeters made a great difference. In metal examination potentials of more than 100,000 volts were frequently employed, and for the present further perfection of the apparatus and photographic methods seemed to be the chief problem. For high penetration induction coils yielding 6 milliamperes at 150,000 volts with a spark gap of 30 centimeters seemed to be the most suitable; under 120,000 volts, 4 milliamperes were sufficient. The mercury-jet interrupters were better worked with hydrogen than with coal gas, especially for heavy currents. Where high generation was not essential, transformer and rotating rectifier-discs, the latter for 106,000 volts were used; in other cases either two valve tubes or one kenotron were recommended.

As regards the photographic side, reinforcement of the effect and elimination of all superfluous rays had chiefly to be studied. With the aid of one intensifying screen the degree of intensification could be raised 15 times; by placing the photographic film between two screens, they had determined differences in thickness of 0.1 millimeter through 45 millimeters of steel. Another advantage secured by these means was that the grain of the crystals in the intensifier—a feature we mentioned last week—was suppressed on the plate; the uneven nature of the crystals created a uniform field. Excellent results had been obtained with films sensitized on both sides. The film had the distinct advantage over the plate that it was flexible.

Fig. 1, annexed, is a reproduction of a radiograph of the entire cylinder of an aeroplane motor, which was obtained by placing four films inside the cylinder; the picture is a positive and the dark parts indicate greater wall thickness. The film could also be mounted in the following way: A base was formed of some polished or silvered metal; on this surface the reinforcing screen was placed, the phosphorescent metal on the top so as to be in contact with the film itself resting on it; on the film the second screen was laid, and on this screen finally the specimen was placed. In order to shut out stray radiations, the specimen was all surrounded with lead, 5 millimeters in thickness. When the specimen was irregular in shape, very fine lead shot was used, which also covered the photographic plate to a depth of at least 15 millimeters or twenty millimeters. This shot should, of course, not be allowed to get between the specimen and the plate, and to prevent that, the specimen was embedded in transparent wax (or plasticine), the wax being trimmed off so that the object rested properly on the plate, on which the lead shot could then be piled; the photographic plate itself was placed over a sheet of lead. Mr. Pearce exhibited a radiograph of a jagged piece of ore-bearing rock obtained in this way, an application of X-rays which deserves attention. That the observation of these precautions repaid itself was evidenced by the radiograph of the base of a 75 millimeter shell; this base looked quite uniform, but there was a faint white spot near the center, marking an invisible weakness in the metal which the pressure test had not revealed.

Mr. Pearce then explained with the aid of slides the arrangements adopted for the protection of the operator. In Messrs. Schneider's works at Harfleur (near Le Hâvre) the operator stood in a cabin, the walls of which were lined with 5 millimeters of lead, safe from the radiations which would spread through the whole room, but able to watch the X-ray bulb through 10 millimeters of anti-X-ray glass or with the aid of small lateral windows and mirrors. When working at more than 80,000 volts, a lead shutter was pulled down in front of the window, or the bulb was placed within a lead-lined box, a mirror being again used for watching the operations; this was the arrangement made for the examination of carbon electrodes and brushes by the Compagnie de Charbons de Nanterre.

We reproduce some of the other interesting slides shown by Mr. Pearce. Though neither the atomic weights of iron (55.8) and nickel (58.7), nor the densities (iron 7.88, nickel 8.9) differ much, Fig. 2 marks strong differences in the shadow depth of this radiograph; the specimen was prepared by embedding a nickel core in iron and planing the two metals down to the same thickness. In an aluminum gear case a defect had been stopped by aid of solder; the flaw came out as a dark spot. Fig. 3 shows the two halves of the bearing of an aeroplane motor, consisting of steel lined with white metal, the latter being held in position by the holes made in the steel; these holes are round, but appear elliptical and rather diffused in the one diagram because the radiation was not normal to the surface. The feature to which Mr. Pearce drew attention is, however, the streaking of this diagram; the streaks were not due to any want of surface finish, but to a faulty composition of the white metal. The carburettor of an aeroplane motor, shown in Fig. 4, was defective in operation. We are not sure that we can trace the defect in the petrol-feed canal which the photograph was said to reveal; but the radiograph, taken with an exposure of only 2 seconds and with the carburettor at the large distance of 125 centimeters from the target (in order to avoid distortion) is certainly a remarkable exemplification of the possibilities of stereoscopic radiography. Mr. Pearce mentioned that they had taken radiographs of the internal parts of an explosive body, 25 centimeters in diameter, made of steel, 12 millimeters thick, and had often been called upon to examine shells, fuses and torpedoes, by means of X-rays. The radiograph of the current distributor of the magneto of an aeroplane motor, Fig. 5, is again remarkable though we are afraid that our readers will not be able to distinguish the segments of the commutator, visible on the original, nor the exact nature of the defective contact which was located by the X-rays, and which is marked by an arrow in Fig. 5.

On behalf of Major J. Hall-Edwards, F.R.S.E., Professor Thomas Turner, of Birmingham, then exhibited some striking radiographs of aluminum castings, mentioning that experiments on brass and other alloys were being proceeded with. We hope to reproduce some of these radiographs on an early occasion.

X-RAY EXAMINATION OF AIR-CRAFT TIMBER.

What X-rays can do in the examination of air-craft woodwork, where the choice of unsuitable material, oversight, carelessness or the concealment of apparently slight mistakes may lead to obscure and fatal accidents, was very forcibly demonstrated in a paper by Captain R. Knox, M.D. (hon. secretary of the Röntgen Society), and Major G. W. C. Kaye, R.A.F., D.Sc., on "The Examination of Aircraft Timber by X-Rays." About the middle of 1918, Major Kaye stated, the Aeronautical Inspection Department decided to utilize X-rays for timber inspection. The equipment of the Cancer Hospital was used, and the results were very encour-

aging. Owing to the severe pressure the method had not yet reached the stage of commercial application at the time of the armistice, but the remaining problems were chiefly of equipment, portability and similar matters. Expressing their great indebtedness to Lieutenant Hudson-Davies and also to Mr. G. F. Westlake, Dr. Kaye explained that they had from the outset recognized the need of rapid visual examination by fluorescent screens, reserving photography for recording cases which the screen had shown to be of interest. As all woods were transparent to X-rays, soft tubes, and spark gaps of 1 inch or 2 inches were sufficient; the output of soft tubes being small, and fluorescent screens requiring excitation by rather hard rays, however, rays as hard as feasible, transformers and Coolidge tubes for 15 milliamperes were used in this testing.

Examinations were made both of the material in the rough condition and of the assembled parts. In aeronautical timber the chief defects to be looked for were spiral grain (in spruce and its substitutes), hidden knots, resin pockets, compression shakes, incipient decay, grub holes, and very light woods. As regards differences in density, due to knots, resin, grub holes, etc., no difficulty was experienced; the other defects were more difficult to detect in thick specimens. The rays chiefly brought out the differences in density between the light spring wood and the denser summer growth, *i.e.*, the annual rings; practically that knowledge was useful only in marking localized hard grain which was objectionable for aircraft purposes. In a radiograph of silver spruce, taken tangentially to the rings, the grain-effect was rope-like, the curious criss-cross effect being due to the presence of "figure" caused by the elongated depression on the annual rings repeated from year to year. Other radiographs exhibited exemplified the slant of the fiber and the glue-joints. On the whole, however, the X-rays did not tell the expert much more as to the timber and semi-finished parts than he could discover by visual inspection; but experts were rare, and semi-skilled inspection would be facilitated by the method.

Passing to the examination of finished and assembled parts, Dr. Kaye remarked that the method was particularly useful for the study of parts, not made of solid wood, but constructed on the laminated or box principle. That construction had become necessary when the supply of high-grade timber had been endangered; smaller timber could be utilized, but the difficulties of inspection were much increased. With that construction defects could be concealed, and unscrupulous and careless workmen were aware of the fact. Notices had been displayed in aircraft factories: "A concealed mistake may cost a brave man his life." In spite of grave difficulties of labor and material, British aeroplanes were superior both in quality and numbers to those of any other nation. But there was a tendency to hide mistakes, the grave nature of which was underrated, and the vigilance of the inspectors had prevented many accidents. In most of the important parts, main-plane wing spars (from end to end of each wing); compression struts (between the two spars of a wing), interplane struts; and longerons, cross-struts and engine bearers (making up the fuselage), the composite construction was permissible. But the strut or spar was completely covered with fabric, veneer or ply wood, and visual inspection was ineffective, unless the inspector could manage to stand over the job all the time.

We select a few of the characteristic radiographs exhibited. Fig. 6 shows the end of a hollow box-strut; the internal strengthening block was badly fitted, each of the screws had split the wood, and the work was altogether discreditable and insecure. Poor workmanship is also manifest in the cutting of the internal strengthening block of the hollow main-wing spar of Fig. 7. The side view of a hollow aileron spar showed the

spar to consist of two halves glued together down the center after the sides had been spindled out; the two halves of the block should register accurately and the sides should be equally thick. In reducing the glued spar to finish dimensions workmen were, however, apt, Dr. Kaye explained, to plane away more wood on one side than on the other, thus reducing the strength.

In some other photographs (not reproduced) the X-rays at once revealed that the halves, which should consist of the same wood, had been made of different materials. The laminated spar, shown in front and side view in Fig. 8, was made up of three laminæ glued together; the external appearance was excellent, but the X-rays disclosed large knots and a grub-hole in the middle layer which would have weakened the spar to a dangerous degree. Fig. 9 exemplifies another kind of reckless work. The wooden skid (top of photograph) had been cut off too short to fit into its socket; to make up the length a piece of packing B had been introduced into the space below. In a wing skid like the one shown this would not be a vital matter, but the same practice had been resorted to in interplane struts. The skid socket, Dr. Kaye added, was aluminum, in. thick, and the radiograph was taken with a current of 10 milliamperes and a spark gap of 8 inches. A similar case was more serious. The internal compression strut did not enter its steel socket (1/20 inch thick) properly and had left an empty space; externally nothing indicated a defect, but the effect of the bad fitting would be that under vibration the strut would gradually work its way into the bottom of the socket, and the straining wires would become slack. Other radiographs illustrate a common defect. When the holes for bolts, crossing parts to be assembled in various directions, are not carefully bored, two bolts may foul one another; the detection of one such fault led to the rejection of dozens of completely finished main-wing planes. The defective riveting and soldering of the end of a steel petrol tank of an aeroplane (Fig. 10) might have attracted notice; but other cases, which Major Kaye illustrated, rivets of a tank which had heads on the outside only (none inside), and cracks in split longerons, cleverly hidden by gluing a shaving over the sand-papered surface, would have escaped notice without the X-rays. The latter have also been of great assistance in tracing the causes of accidents.

In concluding our notice of the general discussion on "The Examination of Materials by X-Rays," held by the Faraday Society and the Röntgen Society on April 29, we pass to contributions affecting the photographic problems of X-ray examination, and we mention in the first instance the communication on "A Method of Testing an X-Ray Tube for Definition," from the Hadfield Research Laboratory in Sheffield, presented by Mr. J. Brooksbank, B.Sc., A.R.C.S. The focal spot, Mr. Brooksbank pointed out, is never a point, but occupies a certain area, and the shadow can hence not be very sharply defined. In metal work the selection of a tube of very fine focus would be of little use, since the effects of secondary radiation would mask the extreme sharpness of the shadow produced in the radiograph. Moreover, the life of a target was generally longer with larger focal spots. Definition must also be taken into account, and Mr. Brooksbank described how this could be effected. The size of the focal spot could sometimes be ascertained by visual inspection; but with hot cathode tubes that was almost impossible. The size of the pitted area of the target roughly indicated the size of the spot. Coolidge had obtained a radiograph of the spot by the pin-hole method. An idea of the definition could be obtained by taking a radiograph of a wire gauze of fine mesh (45 to the inch) at a distance of about 50 centimeters from the spot; the spacings of the wires were irregular, however, and with elliptical focal spots the shadows might be distinct in one direction, but

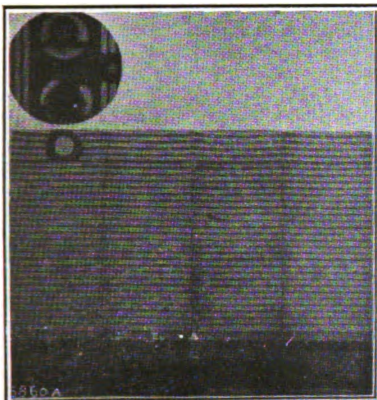


FIG. 1.

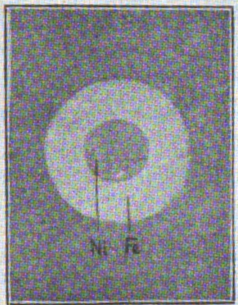


FIG. 2.

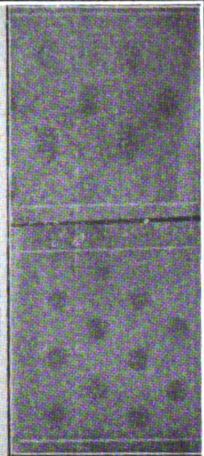


FIG. 3.

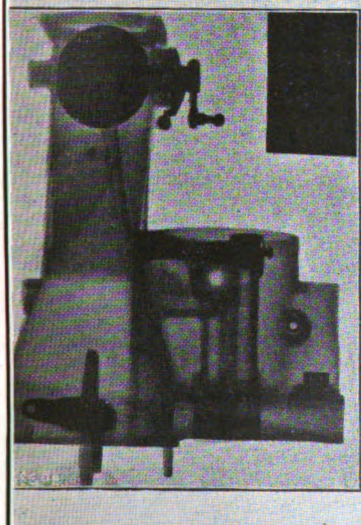


FIG. 4.

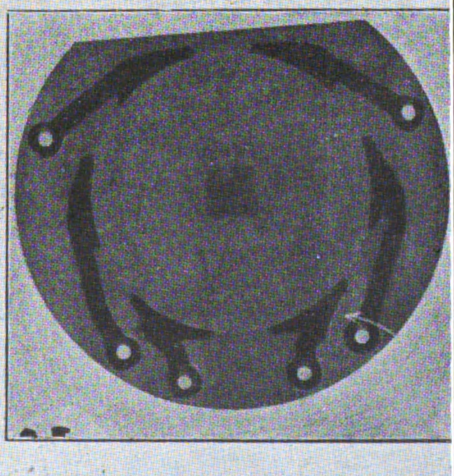


FIG. 5.

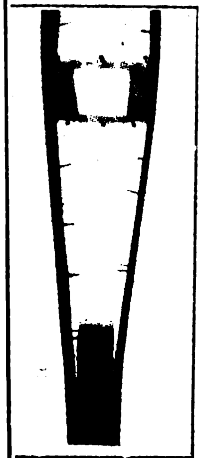


FIG. 6.

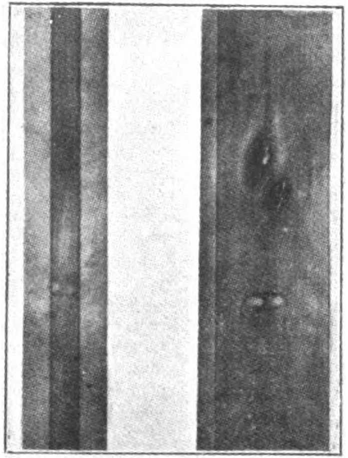


FIG. 8.

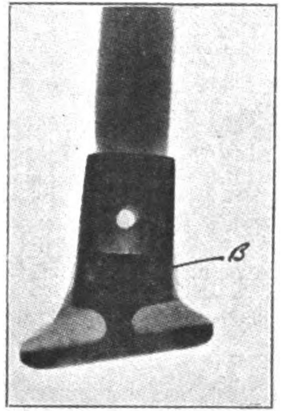
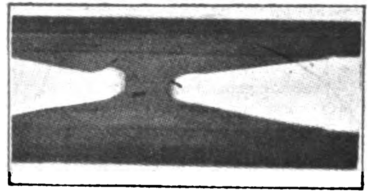


FIG. 9.

FIG. 7.



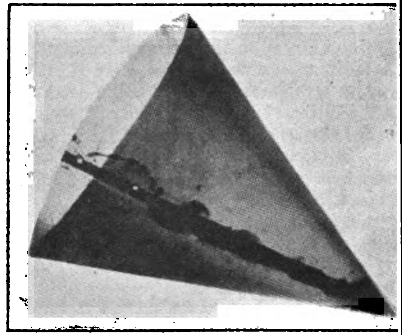


FIG. 10.

not at right angles to that direction. Moreover, sometimes reversal effects were observed. They had therefore used two uniform wires mounted parallel to one another in the same horizontal plane and to the plate; the distance between plate and focal spot was kept at 50 centimeters, and the wires were, in taking a series of radiographs, moved further and further away from the plate. As this distance increased, the shadows become less distinct until, in the "critical" position, the wires could not be seen on the radiograph; that critical distance was adopted as a measure of the definition of the tube.

In his research Mr. Brooksbank studied the shadows given by one wire or by two wires with two sources of X-rays, line sources and circular sources, and determined the intensity of the radiations produced when the distance of the plate from the wires was changed. As each wire received illumination both normally and from the right and left (as would be the case with any source of light of some area, not merely a point) the shadows were diffused, and the distance between the most intense shadows changed with the position of the wires. Comparative tests were made with a Nernst lamp and two pins. With a circular source the critical position was more difficult to decide than with a line source; yet the position could be determined within 2 centimeters or 3 centimeters; from these measurements the area of the focal spot could be deduced. For practical tests Mr. Brooksbank proposd to find by trial the critical position; he would then increase the distance of the wires from the plate slightly, and take radiographs; if the two shadows were separated by a strip of greater intensity than the shadows, the tube satisfied the requirements. Exposures of a few seconds would suffice; the definition would come out worst when the wires were at right angles to the line joining the target and cathode of the tube, and the examination might hence be made with the wires in that position.

Major C. E. S. Phillips then demonstrated how they had put this method to practical use in the War Office X-Ray Laboratory. The apparatus exhibited could hardly be more simple. There is a plate carrier into which the plate is pushed from the side; the plate is covered by a screen which is provided with a circular window. A ring frame attached to a stand holds the two wires, really fine needles, 0.2 millimeters apart; with the aid of a kind of a sliding cam the frame is fixed in one of three positions, at 15 centimeters, 10 centimeters, or 5 centimeters above the window. Radiographs are taken in these three positions, and the wire shadows will come out more or less sharp. Series of three radiographs were exhibited to exemplify how the apparatus is used for estimating the definition; with gas tubes, Major Phillips mentioned, the wires generally were not visible at all on the radiographs.

PHOTOGRAPHIC PLATES, EXPOSURE AND CONTRAST.

The two papers next read exemplified the nature of the photographic problems and the difficulties attending the interpretation of the radiographs. The first, on "The Behavior of Photographic Plates to X-Rays," by Messrs. O. Bloch, F.I.C., and F. F. Renwick, A.C.G.I., F.I.C., of the Ilford Research Laboratory, was presented by Mr. Renwick, who gave a brief account of researches, extending over several years, on X-rays and dry gelatin plates. When the mixed X-rays acted on a plate, he said, the relation between the amount of silver reduced by subsequent development and the quantity of radiation was represented by a curve of the kind of A, Fig. 11; the silver deposits being measured photometrically by determining their opacity logarithm or density ($\log_{10} O = D$); this quantity was plotted against the logarithm of the time of exposure to a

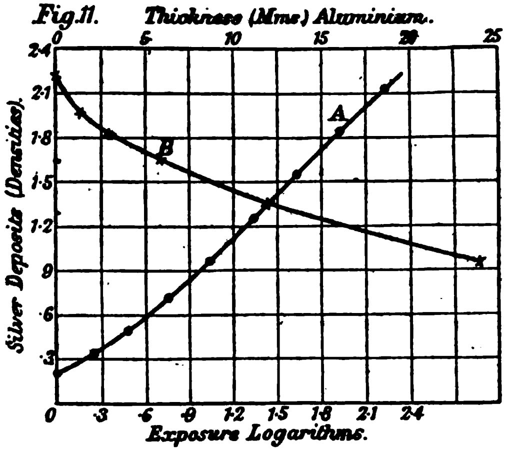


Fig.11. Thickness (Mms) Aluminium.
0 5 10 15 20 25
Silver Deposits (Densities).
2·4 2·1 1·8 1·5 1·2 ·9 ·6 ·3
A
B
0 ·3 ·6 ·9 1·2 1·5 1·8 2·1 2·4
Exposure Logarithms.

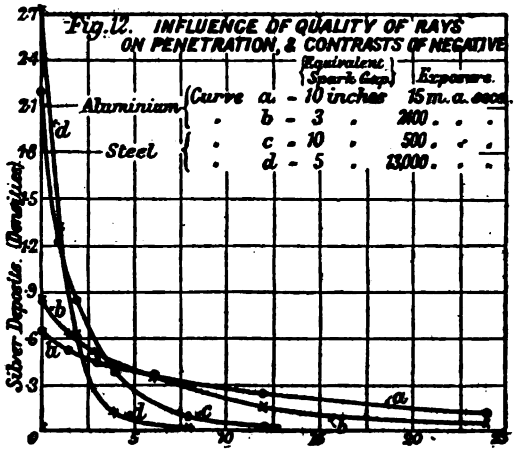


Fig.12. INFLUENCE OF QUALITY OF RAYS ON PENETRATION, & CONTRASTS OF NEGATIVE
Equivalent Spark Gap
Exposure.
Aluminium Curve a. - 10 inches 15 m.a. secs.
b - 3 2400
Steel c - 10 500
d - 5 13,000
Silver Deposits. (Densities)
2·7 2·4 2·1 1·8 1·5 1·2 ·9 ·6 ·3 0
a b c d
5 10 15 20 25

constant radiation. Such X-ray curves had a more pronounced foot (curved lower portion) than in ordinary photography; densities, moreover, were in this foot not proportional to the exposure, as was the case with light in the period of under-exposure. A satisfactory formula, independent of time of development, for expressing the sensitiveness of a plate to X-rays, such as Hurter and Driffield had given for light effect, had not yet been evolved; classification of plates relying on the straight line portion of curve (ignoring the foot) were undesirable as bearing little relation to practice, whilst speed numbers depending entirely upon the foot would possess a real value. Those points were not always heeded, however. When the rays first passed through some material before reaching the plate, selective absorption of the longer waves took place, and the thicker the specimen, the shorter would be the arriving wave-length; there was, moreover, scattering and transformation of the energy into characteristic radiation. The curve connecting the mass of silver reduced with the thickness of material penetrated differed hence from that found for varying the time of exposure to the same radiation. Curve B, Fig. 11, gave such a curve for aluminum. After a fair thickness had been penetrated the curve became inversely proportional to the thickness; to reach that shape early, however, *i.e.*, at moderate thickness, very short waves had to be adopted for metals of high atomic weight. Much depended upon the developer; on curves exhibited metolhydroquinone sodium-carbonate proved a faster developer than pyro-soda, but the increase in rapidity was not the same on different types of plates. The duration of the development did not affect the character of the curve.

The influence of the hardness of the rays was shown in Fig. 12 for a series of steps of aluminum (*a, b*) and of steel (*c, d*), *a* and *c* representing hard rays, *b* and *d* soft rays. The spark gaps used were of 10 inches, 5 inches, 3 inches, respectively, and the currents of 3 milliamperes, 6 milliamperes and 9 milliamperes; the exposures (in milliampere-seconds) are marked on the diagrams. The curves well indicated also that a far greater range of contrast could be obtained with steel than with aluminum; in the case of steel a wide range of intensity should evidently be considered to obtain the full characteristic curve; with the exposures used only the foot had been utilized. To get good contrasts with thick specimens of dense materials both the exposures to hard rays and the developments should be of long duration. Even with the hardest Coolidge tubes (spark gap 15 inches), however, and with very long exposures, 2 inches of steel seemed to be the limit of penetration, and they had hence tried to improve the sensitive materials, turning first to fluorescent intensifying screens. In Figs. 13 and 14 the effects of using commercial calcium tungstate screens were illustrated, the screen being in contact with a plate exposed to increasing doses of radiation; three different plates and various thicknesses of steel were used in Fig. 14. The shape of the curve varied with the ratio: sensitiveness to light to sensitiveness to X-rays, and it would be seen that, in Fig. 14, curve A (slow to light) was convex, B (fast) fairly straight, and C (very fast) concave to the ordinate. Fluorescent intensifier screens not only reduced the time of exposure, but they also increased the contrast, as Fig. 15 further showed. In normal X-ray work (and to a lesser degree also with intensifying screens) the silver deposits produced under the thicker parts of a dense object represented exposures falling within the foot of the curve. The tangent to that curve at any point was a measure of the degree of contrast at that point, and it was obviously not possible to secure good contrasts throughout a negative, unless the quantities of radiation were all sufficiently great to reach values well above those comprised with the shallow part of the foot. Failure to comply with this requirement was the chief reason of the difficulty of

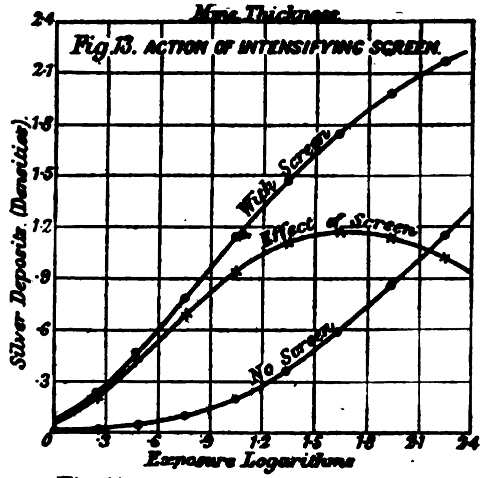
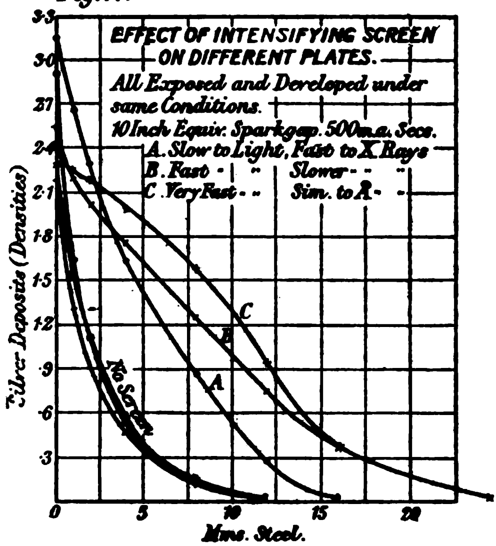

Mms. Thickness.
Fig. 13. ACTION OF INTENSIFYING SCREEN.
With Screen
Effect of Screen
No Screen
Silver Deposits. (Densities).
Exposure Logarithms
0
·3
·6
·9
1·2
1·5
1·8
2·1
2·4

Fig. 14.


EFFECT OF INTENSIFYING SCREEN
ON DIFFERENT PLATES.
All Exposed and Developed under same Conditions.
10 Inch Equiv. Sparkgap. 500 m.a. Secs.
A. Slow to Light, Fast to X Rays
B. Fast " " Slower " "
C. Very Fast " " Sim. to A " "
A
B
C
No Screen
Silver Deposits (Densities)
Mms. Steel.
0
5
10
15
20

bringing out very shallow defects in a thick piece of shell. Unfortunately, even the best commercial intensifying screens were granular and apt to mask fine detail in the radiogram. There was also a very great difficulty about securing good contact between plate and screen, and Mr. Renwick hence doubted the advisability of using separate fluorescent screens in cases where fine definition was essential. Though silver bromide was itself faintly fluorescent to X-rays, there seemed to be little prospect, so far, of making ordinary silver bromide dry plates much more sensitive. Some plates contained fluorescent substances or materials emitting secondary rays under very penetrating X-rays; most of these substances were either injurious to the silver bromide, or were difficult to incorporate in the emulsion. Mr. Renwick thought, however, that success might be achieved.

Dr. R. E. Slade, of the British Photographic Research Association, hardly did himself justice in his paper on "Contrast and Exposure in X-Ray Photography Through Metals." Considering the efficiency of the photographic plate, he said, that the experiments of Barkla and Martyn (1913) were the only ones available showing the relation between photographic effects and wave-length of the radiation (characteristic K radiations of metals), and their curves were not very accurate. Dr. Slade did not add, however, that his very interesting conclusions, which appeared to be little more than mathematical deductions, were in agreement with investigations by Professor Bragg—who had drawn attention to Dr. Slade's paper—and others. The developed negative, Dr. Slade pointed out, left the glass covered with a coating of gelatin containing finely-divided silver. Barkla's curve (Fig. 16) indicated that the absorption due to the silver bromide increased with decreasing wave-length first in a small step, probably due to the bromide, and then in a larger step, due to the silver. If j_o, j_t and j_a represented respectively the intensities of the incident, transmitted and absorbed rays, the ratio j_t / j_o was the transparency (and its reciprocal the opacity), and Dr. Slade showed that when the rays entered the film directly (without having first passed through metal), $j_t = j_o . 10$—D where D was the density of the silver in the film, in the sense of Hurter and Driffield (1890), who meant by density the number of particles of a material spread over unit area, which was a measure of the chemical work done by the light. The density varied with the exposure E (the product of intensity and time) according to $D = A + B \log. E$, where A and B were constants. The two curves, Fig. 17, exemplified the relation between D and log. E for exposures of 8 minutes and 16 minutes. Both were practically straight lines between *b* and *c*; between *a* and *b* was the region of under-exposure, between *c* and *d* the period of over-exposure, and at *c* the exposure would, on an ordinary plate, be about 300 times as long as that at *b*. These curves held in the first instance for ordinary light; Tugman (1915) and Hodgson (1917) had proved that the relation between D and E was the same for X-rays. As it was almost impossible to print from an over-exposed X-ray negative, most X-ray photographs were taken on the under-exposure portion of the curve.

Dr. Slade then considered the case when the X-rays first traversed a thickness of *d* centimeters of a metal of absorption coefficient λ — this designation of the absorption coefficient by λ in problems affecting wave-length is regrettable, but common practice—before entering the film of thickness *t* and absorption coefficient λ_1 (silver bromide). Both λ and λ_1 varied with the wave length; but except at points where there was selective absorption, λ would be equal $a \lambda_1$ (where *a* was a constant), and in that case the equation $\lambda d = 1$ would hold. That meant that the photographic effect for a metal thickness of 2 centimeters would be maximum

if $\lambda = 0.5$, and with most metals the radiations to be used for examination should hence be exceedingly hard (*i.e.*, of short wave-length). The remarkable change of the absorption coefficient λ with wave-length was exemplified by the following table, which gives the λ for aluminum and iron, and for iron also by Fig. 16:—

Wave Length in 10^{-8} *Cm.*	λ *Aluminum.*	*Iron.*
1.95	238	518
1.80	192	527
1.66	140	2460
1.55	128	2100
1.45	106	1740
0.72	8.3	262
0.56	6.7	136
0.50	4.3	86
0.48	3.3	66

In order to attain the greatest contrast, that is, the greatest density difference $D_b - D_a$, where D_a marked the density at any spot where the rays had passed wholly through metal, and D_b at a spot where they had traversed a blowhole, the exposure should be on the correct exposure portion between *b* and *c*, Fig. 17. But with the thick emulsions of X-ray plates the negatives would then be far too dark for printing, and the top of the under-exposure portion should hence be utilized. Since, further, the absorption λ was great for long waves (soft rays), soft rays would give better contrasts; yet we had to use hard rays to shorten the exposures, and Dr. Slade showed that the contrasts were sufficiently great also with hard rays. Finally Dr. Slade referred in particular to the case of iron which [compare the λ curve of Fig. 16 and the table] had very high absorption values for wave-lengths greater than 0.5×10^{-8} cm., thus requiring long exposures. Yet —thickness differences of 0.005 cm. should be quite perceptible, especially, *e.g.*, with rays like the K radiation of nickel, 1.66×10^{-8} cm. As, however, nickel anti-cathodes were not very suitable for photography through iron, anti-cathodes of tungsten or platinum and wave-length of 0.3 cm. or 0.45×10^{-8} cm. were preferable; large spark gaps would be required. In the case of tungsten the K radiation (the only one which mattered for these purposes) had with 68,500 volts a wave-length of 0.18×10^{-8} cm., which became reduced to 0.12 of the same unit at 100,000 volts, thus being fairly steady. In the case of molybdenum the wave-lengths fluctuated very much more.

The important point in these conclusions was that the contrast in photography through, say, 1 cm. of iron, did not depend upon the absolute thickness of the iron, but that the effect was almost entirely due to the hardest rays, though it would be greater with relatively soft rays. That seemed contrary to experience. The explanation was this, Dr. Slade mentioned. Supposing there was a blow hole of 1 mm. in 1 cm. of iron; a good contrast could easily be obtained. If that same bubble were in a specimen 6 inches in thickness, the same contrast could, with the same tube voltage, be obtained by increasing the exposure many thousand times, the function being exponential. In practice we used harder rays to reduce the exposure period; but the contrast was then very much less distinct. We do not know how far these arguments are influenced by the fundamental assumption that the photographic effect depends upon the product: intensity multiplied by time of exposure [milliamperes by seconds), and not upon each of the two factors independently, or that, in other words,

Fig. 15.

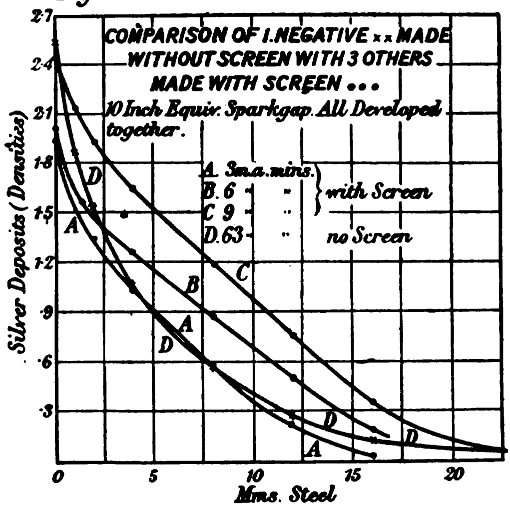


Fig. 16.

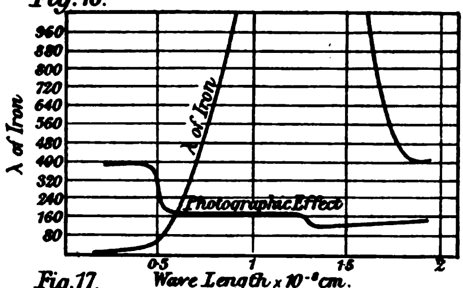
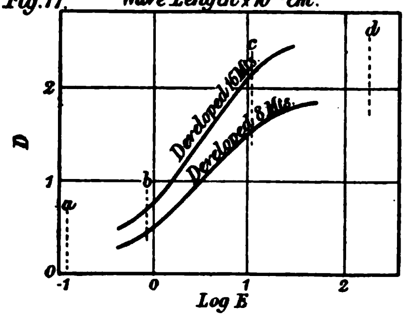


Fig. 17.



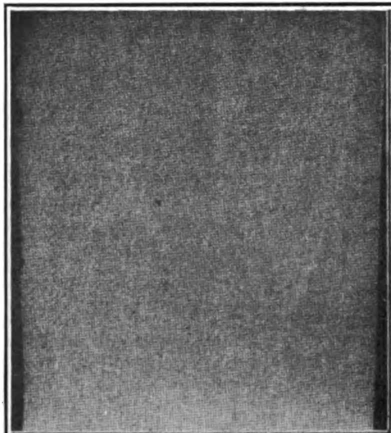


FIG. 18.

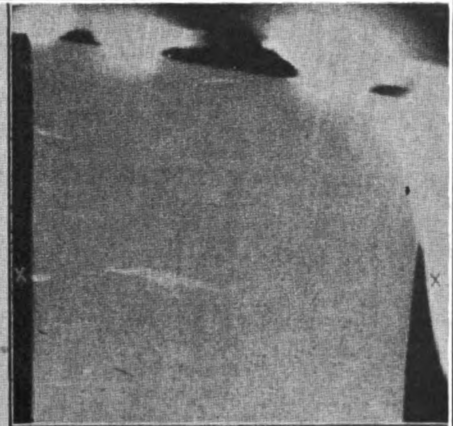


FIG. 19.

FIG. 20.

FIG. 21.

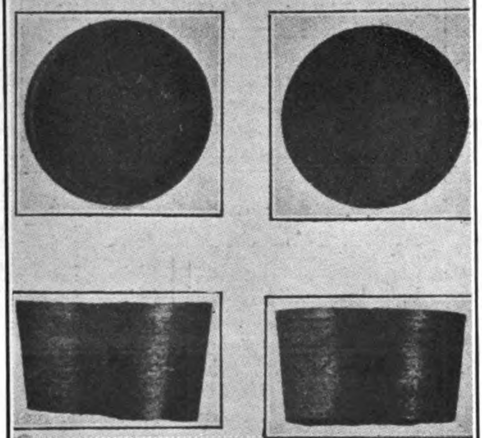


FIG. 22.

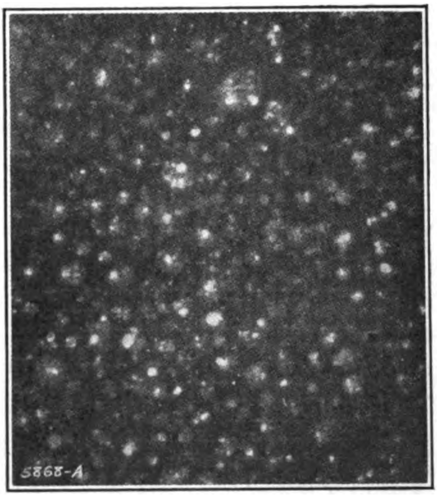


FIG. 23.

weak intensity with long exposure has the same effect as strong intensity with short exposure. Photographers accept that assumption—with reservations—and like them, X-ray workers rely on photographic densitometer tests. But the principle is certainly questionable, especially as regards X-rays. The rays are not homogeneous; they probably fluctuate in hardness from moment to moment and with the hardness varies the scattering which plays an important, yet quantitatively unknown, part. Dr. Slade considers that possible deviations from the milliampere-second rule would hardly affect his conclusions. Mr. Renwick referred to this difficulty in the beginning of his paper, but he expressed exposures in the customary way in milliampere-seconds.

In the general discussion Mr. A. A. Campbell Swinton, F.R.S., remarked that the X-ray bulb problem should be tackled on engineering lines. He did not wish to be limited to milliamperes, but wanted kilowatts. That was not an impossibility. In radiotelegraphy, valves had been used as detectors; now they served as oscillators. The trouble with glass tubes was that strong currents melted them; with steel tubes the occluded gas gave trouble, unless the whole tube were evacuated while red hot. But the interior of the metal tube might be enameled as in Gaede pumps; such tubes, of 2 feet or 3 feet diameter, for 10 kw. or 15 kw., would not cost so very much, and we did not know what they might be able to do. Referring to this, Captain Kaye stated afterwards that Coolidge had, in his especially-cooled tubes, gone up to 14 kw. (20 h.p.) before the war, but the war had forced other work upon him; metal tubes had been introduced by O. Lodge 20 years ago, and all-metal tubes provided with porcelain insulators had been used on the continent by Zehnder and by Siegbahn. Most experimenters seemed, however, to have dropped the metal tube eventually. Captain H. G. Jackson, for some time Ordnance Inspector at Sheffield, said they wanted some means for rapidly examining heavy ordnance forgings made of nickel-chrome steel. That steel showed typical flaws which so far they could only discover during turning; to make fairly sure of having their parts of the required strength, they had to allow for a very large margin of safety. Hence some means for rapidly testing assembled parts would also be most valuable. Mr. S. A. Pollock, of the Postal Department, remarked that the Post Office was very much interested in the possibilities of X-rays; there were many baffling problems—the study of the corrosion and uniformity of cable sheath of lead, of their steel armoring, and of their insulation, among others—which the X-ray tube could help to attack, and they were going to try this method; it had already proved useful in examining gutta-percha for adulterations.

The few photographs, which we finally add, will further exemplify the valuable services that the X-rays can render to the engineer and metallurgist. Figs. 18, 19 and 20 explain the examination by Mr. H. S. Rawdon, of the United States Bureau of Standards, of two pieces of steel, one suspected of being "flaky"; the point was mentioned by Sir Robert Hadfield in his introduction. The latter specimen was cut transversely out of a finished gun forging, of a steel containing 0.37 per cent carbon, 0.01 chromium, 2.85 nickel; a very similar specimen gave a uniformly grey radiograph, the suspected specimen, 3/8-inch thick, showed the flake visible in the line XX. When the specimen was sawn open along this line and broken, the large flake of Fig. 20 (an ordinary photograph) became visible. A similar, very striking case, is illustrated in Figs. 21 and 22, which Professor Bragg exhibited in the course of his address. The photograph Fig. 21 shows a bad flaw in a 5/8-inch turbine casting. The black ring marks the spot where a button was cut out of the specimen. This button, 3/16-inch thick, was photographed in the ordinary way in four views (Fig. 22). The top and bottom of the button do not display any

flaw, but in the two other views, taken from opposite sides, the flaw is very distinct. These X-ray photographs were taken by Dr. Coolidge in the Research Laboratories of the General Electric Company.

Fig 23, from the same source, refers to the improvement of high-conductivity copper effected by boron suboxide. Cast copper is frequently full of pores and blowholes; ordinary deoxidizers added to the copper remove the gases, but alloy with the metal and lower its electric conductivity; the boron suboxide of O. Weintraub deoxidizes the copper without combining with it, and tons of copper are now cast by the General Electric Company with boron. Fig. 23 shows the "unboronized" copper. The radiograph of the "boronized" copper, which Professor Bragg placed next to our photograph, was merely a spotless grey square. Professor Bragg further showed radiographs of some good and some bad welds; they were quite sufficiently plain to let the operator recognize the success or failure of his work, but they might not come out well in our reproduction.

Radiographs of welds and of other specimens and X-ray apparatus, etc., we should not omit to mention, were also exhibited by the British Thomson-Houston Company, by Messrs. H. Cox & Co., and by Messrs. Newton and Wright. One of the exhibits of Messrs. Watson & Sons, to which we have already referred, was their stereo-fluoroscope, an attachment to X-ray outfits, especially designed for the examination of materials and general objects the structure of which appears in stereoscopic relief.—"Engineering."

GERMAN TONNAGE POSITION.

There is as yet no visible sign of a move on the part of the German shipping companies. Reports abound, including one of an amalgamation of the Hamburg American Company and the North German Lloyd, with Herr Heineken as managing director. The Germans say they have lost over 4,000,000 tons out of a total of 4,935,909 tons, and that what is left to them is insufficient for the Baltic trade alone. In any case, for the purpose of international competition, they are and will be for some time to come without ships. They should not, however, be without funds, seeing that the direct financial loss arising from the surrender of their fleets does not fall on them, but on the German Government. Their financial position, however, is declared to be considerably prejudiced by their having to repay their liabilities to the Dutch shipping mortgage banks, which at the present rate of exchange will mean a heavy burden. The amount of this indebtedness can only be surmised, and we are inclined to think it is not great. The German banks were generous supporters of the shipping companies before the war, and their method of furnishing funds was sufficiently satisfactory to render it unnecessary for the latter to look elsewhere for financial help. The banks advanced the money required to build the ships and received the amounts usually written off for depreciation. But whether armed with ample funds or not, and notwithstanding the really terrible account of the condition of the population in Germany described in the parliamentary paper just issued (Cd. 280), the German companies are, there is reason to believe, now busy making preparations in Holland and Scandinavia both in the form of chartering and in acquiring interests in steamship companies.—"Shipbuilding and Shipping Record."

LUBRICATING SYSTEMS FOR GEARED TURBINES.

Probably the most important detail in the laying out of a geared turbine installation is the provision of a simple and reliable oiling system. So vital is the question of lubrication to the operation of the machinery, that too much attention cannot be given to the design and lay-out of the oiling gear fitted. The pressure system of lubrication is always adopted in modern turbine installations, and, in addition to providing for the lubrication of the main bearings of the turbines themselves, provision must also be made for the efficient lubrication of the gears, this comprising the lubrication of the various bearings supporting the pinion shafts and the gear-wheel shafts, in addition to the lubrication of the teeth at their point of contact. This latter is usually performed by a series of jets of oil, delivered from pipes mounted near the point of contact of the pinion and gear teeth, the jets being delivered right on to the point of contact, and in the direction of motion of the teeth, so that there is always a film of lubricant preventing the actual metallic contact of the teeth of the pinion and of the gear. In fact, it may be said that the problem of efficient turbine lubrication consists of the provision and the maintenance of a film of oil at every point where two metallic surfaces move the one over the other.

Two systems of lubrication have been devised, the first in which separate oil pumps are provided, usually in duplicate, and the second, in which the pumps are direct driven from one of the shafts of the gearing, in the case of double-reduction gearing the intermediate gear shaft being employed for this purpose. In both systems the lubricant is delivered to a gravity tank, situated as high as possible above the machinery, 20 feet being considered a minimum figure in this connection, the oil to the bearings and the gears being supplied from this tank by gravity, some form of float valve being provided to automatically govern the delivery from the pumps. In the case of vessels where there is insufficient head-room to fit the gravity tank at a proper height, a pressure oiling system has to be adopted, and special precautions have to be provided to ensure an uninterrupted supply. The overhead gravity tank is usually provided with three strainers, having a successively finer mesh, it being a matter of supreme importance that the lubricant should be absolutely free from dirt before entering the bearings or the gears. In the event of the screens becoming choked, however, there must be no interruption of the service, and hence they are mounted in the gravity tank in large frames, so that the oil can flow over the top of the screens. Moreover, they can be readily removed for cleaning, and any dirt falling from the screen while they are being removed will be caught in the frames, and thus prevented from remaining in the oil. The outlet from the gravity tank to the machinery should be well above the bottom, so as to avoid drawing off any sediment; but a pipe should, of course, be provided at the bottom of the tank for drawing off this oil to a filter or a settling tank. The character of the oil in the bottom of the gravity tank should be regularly observed, so that the engineer on watch may be sure that no water is entering the system from any source.

With the pressure oiling system, as found on board torpedo-boat destroyers, it is still necessary to employ some form of strainer to filter the oil before delivery to the machinery. There is, however, a danger that these may, on becoming choked up, interrupt the service; or if the pumps are sufficiently powerful the strainers may be burst, thus letting dirt through. To avoid this, the strainers by-pass through a spring-loaded check valve, sometimes loaded to about ½ pound per square inch, so that with a slight increase of resistance of the strainers the oil will go via the by-pass, and the service will never be interrupted. As a further means of preventing the circulation of dirt with the lubricant, the suction

to the oil pumps is placed several inches above the bottom of the drain-oil tank, so that any dirt or emulsion in the oil which drains from the bearings and the gears into this tank may settle at the bottom, and not be drawn up and delivered again to the machinery. An additional suction is, however, employed at the bottom of the tank by means of which the dirty oil may be pumped through a filter, and to permit the complete emptying of the tank for cleaning purposes. This drain tank is located sufficiently low to ensure the oil draining freely to it from the machinery, the tank being as large as possible, with connecting pipes of ample size, so that, as far as possible, even with extreme rolling and pitching of the vessel, there will be no leakage of oil from the bearings. The oil pumps are placed near the drain tanks, and at as low a level as possible, in order to reduce the suction head to the least possible. A further detail which has also to be carefully considered is the provision of oil coolers, since the oil, on leaving the machinery, is often at a temperature considerably above that at which it was supplied, and any further rise on again passing through might lead to a considerable falling off in its lubricating properties. The oil coolers are usually arranged on the discharge side of the pumps, between the pumps and the gravity tank. The coolers are duplicated, so that while one set is in use the other may be cut out of service. The oil coolers should also be by-passed with a spring-loaded check valve, loaded to some amount greater than the resistance of the cooler, so that under no conditions can mis-handling of the valves cause interruption of the service.

Opinion amongst turbine designers is divided as to the relative merits of the oil pumps driven direct from the gear shaft, as compared with the independent pump. It is recognized that under circumstances when the vessel is maneuvering for a long time, the direct-driven pumps may not be capable of supplying a sufficient quantity of oil; it is therefore usual to install a small independent oil pump. Moreover, with the direct-driven pump it is certain that a larger quantity of oil is circulated than is actually necessary. The independent pumps, while lacking the extreme reliability of the direct-driven pumps, are, when properly controlled, able to deliver only just that quantity of oil needed by the system, and this, no doubt, will have much to do with increasing the life of the lubricant, as well as tending to simplify the filtering arrangements.—"Shipbuilding and Shipping Record."

SOME DEVELOPMENTS IN AIRCRAFT DESIGN AND APPLICATION DURING THE WAR.*

By the Right Hon. Lord Weir of Eastwood, P. C., Honorary Fellow. (Abridged.)

Any attempt to review the progress or development of aviation during the war, and to confine such a review to the permissible limits of a paper, involves a mere indication of some of the more salient features of the development. Moreover, it is difficult to confine any such review entirely to the scientific or engineering aspect of the problem, on account of the peculiarly close association of the technique of construction with the technique of use. The war development of a technical product, such as an aeroplane, necessarily comprises the rapid embodiment of field experience in the design and production of *materiel,* and this compelling influence during the war largely governed the policy of design, supply and production.

In August, 1914, the British Air Services consisted of a naval wing and

*Paper read before the North-East Coast Institution of Engineers and Shipbuilders, July 10, 1919.

a military wing, the first controlled by the Admiralty, and the second by the War Office. The naval wing, or Royal Naval Air Service as it was termed, possessed a total of 93 machines, the military wing or Royal Flying Corps possessed a total of 179 machines. In October, 1918, just prior to the armistice, the Royal Air Force possessed over 22,000 effective machines.

For the first two years of the war, the supply organizations for aircraft were departments of the War Office and Admiralty, and no distinct technical departments existed. In January, 1917, the second Air Board, under the chairmanship of Viscount Cowdray, succeeded the original Air Board under Viscount Curzon, and was constituted with additional duties and responsibilities. The Board became responsible for the allocation of aeronautical supplies between the two flying services, and for the supervision of design of all aeronautical material, this latter responsibility being vested in the technical department of the Air Board under Brigadier-General Pitcher. Concurrently with this, the entire responsibility for supply and production of all aircraft *materiel* was handed over by the War Office and Admiralty to the Ministry of Munitions. Toward the end of 1917, a movement in favor of concentrating the entire administration of everything relating to war aviation in a single Government department crystallized in the constitution of the Air Ministry under a Secretary of State for Air, the naval and military air services being amalgamated to form the Royal Air Force in April, 1918.

The establishment of the Royal Air Force as an independent fighting force of the Crown has been thoroughly well justified, and the progress with regard to civil aviation in this country since the armistice is largely due to the existence of a single air authority. Concurrently with the institution of the Air Ministry, the necessity of placing design and supply under a single authority was recognized by the constitution, within the Ministry of Munitions, of the Aircraft Production Department, which assumed full responsibility for all questions of design, supply and production.

An appreciation of the progress made in the domain of supply and production can be obtained from the fact that the average monthly delivery of aeroplanes either from British or foreign sources to our flying service during the first twelve months of the war was 50 per month, while, during the last twelve months of the war, the average deliveries were 2,700 per month. The capacity of the facilities in this country for the production of aeroplanes at the date of the armistice was approximately 3,500 complete machines per month. To those associated with marine engineering, it may be of interest to state that the total horsepower of areo engines produced in the last twelve months of the war, approximated to 8,000,000 of brake horsepower, a figure quite comparable with the total horsepower of the marine engine output of the country.

It may be of interest to recite some of the difficulties encountered in this work, although perhaps the more important of these difficulties were inseperable from the industrial position of the country at the period when the effort towards expansion was made:—

1. The lack of highly-skilled labor, in particular that required for engine production, due to the almost complete absorption of such labor by other and earlier war enterprises.
2. The very high standard of material and workmanship involved so that safety might be ensured on the low permissible weight of the product.
3. The necessity of creating and building up entirely new manufacturing facilities and organizations.
4. The grave influence on production of modifications in design shown necessary by field experience, and the necessity for the rapid embodiment of these in the product.

5. The inability to take the fullest advantages of standardization, owing to the necessity of making continuous progress in design and performances of machines.

6. The extreme complexity and variety of the elements contributing to the provision and equipment of war aeroplanes.

For example, the provision of satisfactory timber was a continual difficulty—at times an actual menace to the whole development. The textile problem became very grave when the supplies of Russian flax were cut off, and we were compelled to develop additional sources in Ireland and in the Colonies. Acute difficulties were experienced in connection with the development of the chemicals required for dope manufacture. At other times, the supply of machine guns gave much anxiety, while the development of the synchronizing gear for these guns necessitated very urgent treatment.

The production of ball bearings involved the provision of new facilities on a colossal scale. The magneto supply involved the building up of an entirely new industry, while the manufacture of the numerous classes of instruments, cameras, radiators and other fittings in each case formed a problem by itself.

The solution of these problems from a production point of view constitutes an outstanding example of the enterprise, courage and ingenuity of British industry. Many mistakes were made, but most of the difficulties were solved and many valuable lessons have been learned. It is a matter of great regret that so many of these enterprises, built up for specific war requirements, cannot be maintained under peace conditions. The development of civil aviation will not, for many years, absorb even a fraction of the war facilities, and a large amount of waste cannot be avoided in the reduction of these industries to a peace-time basis.

The constitution in 1917 of the first Technical Department dealing with aircraft design represented one of the most valuable steps in advance of previous organization, and the work of this department very largely contributed to the position of technical supremacy held by this country at the close of the war. One of the factors contributing to this success was undoubtedly the adoption of a policy of giving ample freedom of opportunity to private designers, because in the development of a new art, such as aircraft design, any adherence to a single school of thought is dangerous, and the basis of design and experiment should be broadened as much as possible.

In the earlier days of the development the official Government designs of the Royal Aircraft Factory at Farnborough predominated, but the change in policy should not be taken to represent a reflection on the many valuable designs produced at Farnborough. These designs exercised a great influence on all future designs, while the meticulous care in the details, which was the feature of Farnborough practice, has been wholly useful and valuable in its general influence. To the British aircraft designers as a whole, and in particular to the pioneer designers and manufacturers, the greatest credit is due for their courage, skill and ingenuity.

Such is the briefest possible review of the political and administrative conditions under which the developments in war aviation have been carried out. In dealing with the salient features of progress and design two aspects have been separately treated—the aerodynamic aspect and the applicational aspect. As regards seaplanes it is not proposed to deal with their detail development, as this followed generally on the same lines as the aeroplanes with the special adaptations to meet marine conditions.

PART I.—AERODYNAMICAL ASPECT.

Loading.—The outstanding feature desired in war aeroplane performance was expressed generally as the maximum of speed and climb, and it was

soon perceived that this feature could be best achieved by a reduction of the weight carried per horsepower. The advance in other aerodynamic features such as wing section, reduction of air resistance, etc., was considerable, but could not be compared in importance with the reduction of the weight per horsepower.

At the beginning of the war, loadings were about 23 pounds per horsepower and 4 pounds per square foot of lifting surface; at the close of the war, nearly 7 pounds per horsepower, and 10 pounds per square foot. This is a remarkable result when it is considered that 7 pounds per horsepower represents the power loading of the total weight of the aeroplane, comprising the engine, its petrol and oil, the aeroplane itself, the pilot and all his fighting equipment. The rate of progress achieved in this development entailed a corresponding large demand on the skill of the pilot, but it was found that the pilot's skill always advanced rapidly as the demands upon it.

Wing Sections.—Wing sections steadily developed from the early sections with their hollow undersides and tops of almost circular camber, to the high efficiency sections of the present day with flat undersides bulged slightly downwards for the spars, and upper surfaces with a quick nose curvature, flattened top and easy run aft.

The illustration, Fig. 1, shows these differences fairly well. It may be noted as an interesting fact, that the Germans at no time made use of a high efficiency section such as the R. A. F. 15 shown.

Air Resistance.—No revolutionary advances have been made during the war in regard to the reduction of air resistance, but the general trend has been the more rigid application of previous knowledge; the conversion of piano wire or cable to streamline wires for main bracing is perhaps the most noteworthy feature. Until quite recently stream-line wires were not adopted by any other country than this, although for fast aeroplanes the gain is considerable—approaching 10 miles an hour on a machine with a speed of 110 miles per hour.

The fuselages were made of fairer shape, and pilots and accessories were more carefully enclosed and protected.

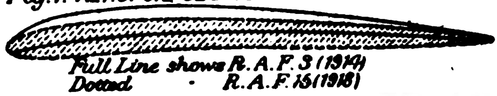
Stresses and Factors of Safety.—At the outbreak of hostilities very little was known of the magnitude of the forces which might occur in flight, and certain more or less arbitrary rules were used in determining the strength of the aeroplane. Many possible methods of failure were never considered, and in some cases it was only through the light shed by certain accidents that progress was made. Certain definite forms of failure which are now always considered and guarded against were only discovered after long and careful examination and analysis of accidents.

When an aeroplane is flying level at a constant speed the air loads acting vertically on the lifting surfaces (wings and tail) must, of course, be equal to the total weight of the aeroplane. Other conditions must be simultaneously fulfilled, but this is the primary one.

This air load on the wings is known as the normal or unit flight load and is the basis of all strength calculations. It will be clear that in order to accelerate the aeroplane, extra forces must be applied, to it, and so in many maneuvers through which the aeroplane is put, extra loads are thrown upon the structure.

These forces are measured in terms of the unit flight load, and their magnitudes are determined either by calculation or by experiment. Both methods are used, and the advance in knowledge on this question, during the war, has been one of the principal causes of the elimination of practically all accidents which may be genuinely attributed to structural failure, and the great reduction in the percentage weight of aeroplane structures. The experimental method of determining the values of these loads consisted in carrying an accelerometer on an aeroplane. This instrument

Fig. 1. AEROFOIL SECTIONS



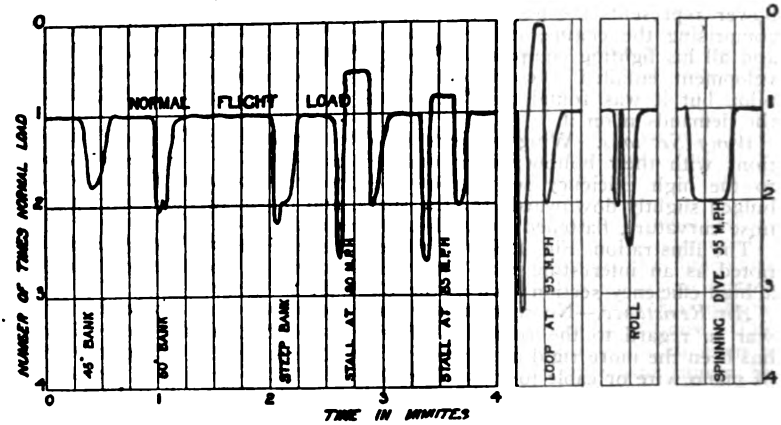


FIG. 2. ACCELEROMETER TESTS. FLIGHT LOADS B.E. 2C.

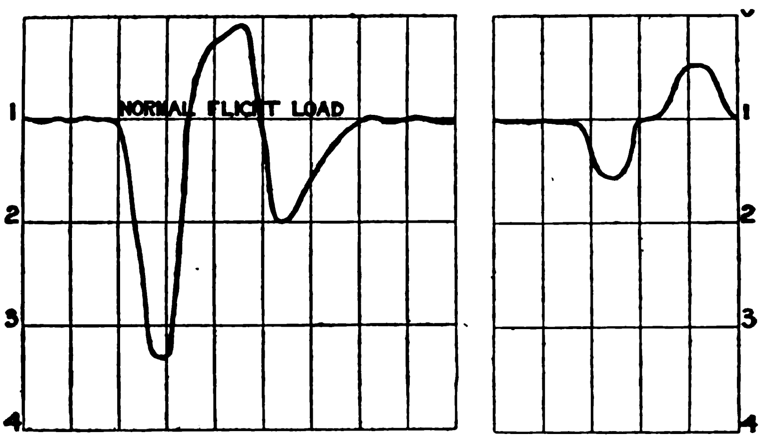


FIG. 3. ACCELEROMETER TESTS. LOADS ON B.E. 2C.

was self-recording and registered the amount of the accelerations given to the craft during various maneuvers. Developed by Professor Lindemann and Dr. Searle at the R. A. E., its value has been considerable. The illustrations (Figs. 2 and 3) were obtained from one of their autographic records, and these and the attached table will give some idea of the value of the loads which occur during aerial maneuvers. It must be understood that these loads are not the maxima which could occur, but only the maxima registered during the experiments.

Table of Loads.

Ordinary bumpy weather.	1½ to ½ normal load.
Spiral dive at 70 miles per hour.	2.8 times normal load.
In a mock fight between S.E. 5 A. and R.E. 8.	3 times normal load was common; at times this increased to nearly 4.

Until all the loads which can possibly occur on an aeroplane are known it is impossible to design an absolutely scientific structure, but considerable progress has been made during the war in this direction, and it is at least possible now to design a structure that will not break in the air except under certain conditions against which the pilot can be warned.

In all other branches of engineering the prevention of failure is obtained by the introduction of a factor of safety, so that the highest possible loading which can occur in the structure stresses the material to only a fraction of its ultimate stress.

With an aeroplane this method is impossible on account of weight limitations, and in consequence a method is adopted of specifying load factors instead of factors of safety.

This method consists of determining—largely as a result of previous experience—what extra loads any particular type of aeroplane is likely to meet under the conditions of flight for which it is intended, and the structure is designed with just sufficient strength to carry these loads, that is to say, when these loads occur the material is stressed either up to its elastic or its ultimate stress. In the usual engineering sense, therefore, the factor of safety under this condition of flight is only one, while the load factor may be from four to seven, *i.e.*, the loads which cause failure are from four to seven times normal flight loads as specified previously.

In aeroplanes of the scout class it has been customary to ask for a load factor of seven, but this only corresponds to a factor of safety of about 0.6. That is to say, the aeroplane by certain maneuvers could be broken in the air. This fact is well known to all pilots, and such a breakage now seldom occurs. It will be seen from the above table that in ordinary bumpy weather the loads were only increased about 1½ times normal.

Since all other increased loads arise from "stunts" it would appear that commercial aeroplanes will seldom have to contend with the heavy loads met with in war flying, and it may be possible on this account to cut down load factors especially on heavier types. The illustration shows the difference between the loads in a loop and loads due to gusty weather upon the same aeroplane.

Analysis of Weights.—As in all forms of science, progress is very largely dependent on the ability to measure accurately the various items and factors. No genius can make up for the neglect of this essential. Once all the factors, whether they be weights, sizes or performances, are accurately measured, it is not so difficult to decide in which direction to proceed. A large amount of attention was given to the careful analysis of the weights of all available aeroplanes, good or bad, and tables were prepared of which samples are given in Figs. 4, 5 and 6.

ENGINE: 250 ROLLS-ROYCE Mk. III.			
TYPE	WATER COOLED V 60°		
NORMAL B.H.P.	270	R.P.M.	1600
MAX -	286	R.P.M.	1850
Nº FITTED	ONE		
AIRSCREW R.P.M. ÷ ENGINE R.P.M.	·64		
FUEL PER NORMAL B.H.P. HOUR	·5706 lbs.		
OIL - - - -	·0254 lbs.		
FUEL PER MAX B.H.P. HOUR	-		
OIL - - - -	-		

AEROPLANE: DE HAVILLAND 4.			
TYPE	TWO SEATER-TRACTOR		
Nº OF WINGS	BIPLANE		
TOP WING SPAN	42' 6"	CHORD	
BOTTOM - -	42' 6"		-
MIDDLE - -	-		-
OVERALL LENGTH	30' 8"	HEIGHT	10' 5"
GAP, TOP TO BOTTOM		TOP TO MID.	
GAP ÷ CHORD		DIHEDRAL	
DIST. BET. LEAD'G EDGES of LOWER WING & T.P.			

		AREAS ☐'		WEIGHTS LBS.	WT/SQ FT.	% WEIGHT
STRUCTURE	WINGS	224	TOP PLANE	199	·9	
		212	BOTTOM PLANE	186	·9	
			MIDDLE PLANE	-		
			AILERON PULLEYS	5		
			STRUTS (Nº =)	52		
			EXTERNAL BRACING WIRES	24		
		436	TOTAL WINGS.	466	1·1	14·1
	TAIL.	38	TAIL PLANES	17	·5	
		24	ELEVATORS	17	·7	
		4·25	FINS	4	1·0	
		13	RUDDERS	11	·9	
		79	TOTAL TAIL.	49	·6	1·4
	BODY		FUSELAGE	*425		
			CHASSIS	100		
			TAIL SKID	8		
			CONTROLS (included in Fuselage)	-		
			TOTAL BODY.	533	1·2	16·1
			TOTAL WEIGHT OF STRUCTURE UNIT.	1048	2·4	31·6
					WT/H.P.	
POWER	PLANT.		ENGINE DRY	880	3·3	
			PROPELLER	70	2·6	
		GALLS 8	RADIATOR & PIPING & WATER	*160	·6	
			ENGINE ACCESSORIES	*10		
			POWER UNIT. EXCLUDING FUEL, OIL, TANKS	1120	4·2	33·8
	SUPPLIES.	GALLS 59	FUEL TANKS & PIPING	*66		
		" 4½	OIL TANKS & PIPING	*14		
		HOURS	FUEL	425	1·6	
		"	OIL	40	·15	
			TOTAL WEIGHT OF POWER UNIT.	1665	6·2	50·3
LOAD	HUMAN		CREW	360		
	RECONNAISSANCE		INSTRUMENTS	28		
			CAMERA	27		
			W.T.			
			SUNDRIES			
	ARMAMENT		GUNS & AMMUNITION	185		
			BOMBS & GEAR	-		
			ARMOUR	-		
			TOTAL WEIGHT OF LOAD UNIT.	600	2·2	18·1
AIR BOARD T.S.D.			TOTAL WEIGHT OF MACHINE.	3313	7·6 per ☐' / 12·3 per HP	100·0

* approximate

FIG. 4. ANALYSIS OF WEIGHT. DE HAVILLAND 4.

ENGINE: 130 CLERGET		AEROPLANE: SOPWITH CAMEL, F 1/3.	
TYPE	AIR-COOLED ROTARY	TYPE	SINGLE SEATER FIGHTER
NORMAL B.H.P.	127·75 R.P.M. 1250	No OF WINGS	BIPLANE
MAX.	- R.P.M.	TOP WING SPAN	28' CHORD
No FITTED	ONE	BOTTOM -	28' -
AIRSCREW R.P.M. ÷ ENGINE R.P.M.	1.	MIDDLE -	- - -
FUEL PER NORMAL B.H.P. HOUR	·5715 lbs	OVERALL LENGTH	18' 8" – HEIGHT 8' 6"
OIL	·1563 lbs.	GAP, TOP TO BOTTOM	TOP TO MID.
FUEL PER MAX. B.H.P. HOUR		GAP ÷ CHORD	DIHEDRAL
OIL		DIST. BET LEADg EDGES of LOWER WING & T.P.	

	AREAS ☐'			WEIGHTS LBS.	WT/SQ. FT.	% WEIGHT
STRUCTURE	121	WINGS	TOP PLANE	104	·9	
	110		BOTTOM PLANE	90	·8	
			MIDDLE PLANE	-		
			STRUTS (No = 8)	15		
			EXTERNAL BRACING WIRES	20		
	231		TOTAL WINGS.	229	1·0	15·75
	14	TAIL	TAIL PLANES	13	·9	
	10·5		ELEVATORS	8	·8	
	3		FINS	2	·8	
	4·9		RUDDERS	3·5	·7	
	32·4		TOTAL TAIL.	27	·8	1·85
		BODY	FUSELAGE	108		7·4
			CHASSIS	70		4·8
			TAIL SKID	3		
			CONTROLS	14		·9
			TOTAL BODY.	195	·845	13·4
			TOTAL WEIGHT OF STRUCTURE UNIT.	451·	1·95	31·0
POWER		PLANT	ENGINE DRY	375	WT/H.P. 2·9	
			PROPELLER	30	·2	
	GALLS.		RADIATOR & PIPING & WATER			
			ENGINE ACCESSORIES	27	·2	
			POWER UNIT, EXCLUDING FUEL, OIL, TANKS	432	3·4	29·7
	GALLS. 26	SUPPLIES	FUEL TANKS	24	9 lb per gal	
	- 6		OIL TANKS & PIPING (PIPES FOR FUEL & OIL 6½)	13	1 lb per gal	
	HOURS 2½		FUEL	180	1·4	
	-		OIL	63	·5	
			TOTAL WEIGHT OF POWER UNIT.	712·	5·6	49·0
LOAD		HUMAN	CREW	180		13·0
			INSTRUMENTS	10		
		RECONNAISSANCE	CAMERA	-		
			W.T.	-		
			SUNDRIES	-		
		ARMAMENT	GUNS & AMMUNITION	101		7·
			BOMBS & GEAR	-		
			ARMOUR	-		
			TOTAL WEIGHT OF LOAD UNIT.	291	2·3	20·0
AIR BOARD TECH. DEPT.			TOTAL WEIGHT OF MACHINE.	1454	6·3 per ☐' / 11·4 H.P.	100·0

FIG. 5. ANALYSIS OF WEIGHT. SOPWITH-CAMEL.

Engine: Rolls-Royce	Aeroplane: Handley-Page
Type Eagle 8	Type 0 400 Twin Tractor
Normal B.H.P. 359 R.P.M. 1800	No. of Wings Biplane
Max. " 368 R.P.M. 1900	Top Wing Span 100' 0" Chord 10' 0"
No. fitted Two	Bottom " 70' 0" " 10' 0"
Airscrew R.P.M. ÷ Engine R.P.M. ·6	
Fuel per Normal B.H.P. Hour ·50 lbs.	Overall Length 62' 10¼" Height 22' 0"
Oil " " " " ·025 lbs.	Gap, Top to Bottom 11' 0"
Fuel per Max. B.H.P. Hour	Gap ÷ Chord 1·1 Dihedral 4°
Oil " " " "	Dist. bet. Lead. Edge of Lower Wing & Elev. Hinge = 46

	Areas □'			Weights Lbs.	Wt./Sq. Ft.	% Weight
Structure	1023	Wings.	Top Plane	944	·9	7·2
	626		Bottom Plane	578	·9	4·4
			Footplates, Hinges &c.	159		1·2
			Struts (No. = 14)	226		1·7
			External Bracing Wires	258		1·9
	1649		Total Wings.	2165	1·3	16·4
	123·5	Tail.	2 Tail Planes	72	·6	·6
	65·3		2 Elevators	44	·7	·3
	14·7		1 Fin	7·5	·5	·3
	45·7		2 Rudders	35·5	·8	
			6 Tail Plane Struts & Bracing Wires	39		·3
			Total Tail.	198		1·5
		Body.	Fuselage	1425		10·8
			Chassis	724		5·5
			Tail Skid	36		·6
			Controls	52		
			Total Body.	2237		16·9
	Total Weight of Structure Unit.			4600		34·8
Power		Plant.	2 Engines Dry with starting mags and exhaust manifolds	1945	Wt./H.P. 2·7	14·8
			2 Propellers	121	·2	·9
	Galls. 20		2 Radiators & Piping & Water	506	·7	3·8
			Engine Accessories incl. bearers and cowling	375	·5	2·8
			Power Unit, excluding fuel, oil, tanks	2947	4·1	22·3
	Galls. 298	Supplies.	Fuel Tanks & Piping & Pumps	261	·5	2·5
	" 30		Oil Tanks & Piping	67		
	Hours 6 at G.L.		Fuel	2150	3·0	16·3
	" 8 at G.L.		Oil	290	·4	2·2
	Total Weight of Power Unit.			5715	8·0	43·3
Load.	Human		Crew	540		4·1
			Instruments	35		·2
	Reconnaissance		Camera	"		
			W.T.	"		
			Sundries	155		17·6 { 1·2
	Armament		Guns & Ammunition, 4 Lewis 66, Amm. 108, Mounts 61	235		1·8
			Bombs & Gear 16-112 lb. bombs = 1800, Gear 120	1920		14·6
			Armour			
	Total Weight of Load Unit			2885		21·9
Tech. Dep. A.P.	Total Weight of Machine.			13200	8·0 per □' / 18·4 H.P.	100·0

Fig. 6. Analysis of Weight. Handley-Page 0/400.

By a study and comparison of these results in conjunction with the strength of the various aeroplanes, much knowledge of the possibilities of construction was obtained. The interesting fact was arrived at—that for a range of well-designed practical war types, the structural percentage remains roughly constant for aeroplanes of total weight varying from 1,000 pounds to 30,000 pounds.

From a theoretical point of view, this is somewhat surprising because, as is well known to engineers, the law of dimensions lays down that area increases as the square and weight as the cube of the dimensions, and that this will therefore put a limit on size. In fact, a very eminent aerodynamic theorist, working on these lines several years ago, put the limit of the weight of an aeroplane at about 10,000 pounds. There is very little doubt now that aeroplanes of 100,000 pounds are a practical proposition.

Some of the reasons for this apparent theoretical discrepancy may be of interest. One is, that the larger the aeroplane, the more sober is the method of progression. No one wants to loop or do vertical banks on a big passenger aeroplane, and therefore it is not necessary to maintain as high a strength factor on the big type as on the small. As far as actual flying stresses are concerned, it would be possible for an aeroplane with a load factor of only 1½ to be flown without collapse even on a windy day. This statement is only used for illustration, and must not be taken to suggest that a load factor of 1½ would produce a practical aeroplane; it would probably be too weak to stand landing. Another reason for the discrepancy is, that the bigger the aeroplane the more detailed can be the design work, and it becomes possible to use material in a more efficient way.

It is not considered probable that aeroplanes made of wood will increase to a size representing a weight of much more than 40,000 pounds, but by the use of high-grade steel and duralumin, it will certainly be possible to go far beyond this limit.

Influence of Tunnel Experiments.—At one time the small scale work carried out in the wind tunnel was regarded as of little practical value; now, judging by the results, I do not think that it would be too much to say that the work which was put into tunnel research, when this work at the National Physical Laboratory was under the direction of Mr. Leonard Bairstow, before as well as during the war, was the real basis of the technical success which we undoubtedly attained, particularly in the aerodynamic field. The data from such work is more useful to the designer from a comparative aspect than for the absolute values obtained, but without their help it is only too easy to stray off into blind alleys leading nowhere except to disappointment.

Some designers have undoubtedly a wonderful facility in guessing the next step to take, but they are too few and far between for responsible authorities to rely solely upon them. In any case such men always work best if there is a solid background of research knowledge behind them, from which they draw sometimes perhaps unconsciously. In 1914 no private firms had a wind tunnel of their own or went in seriously for research. Now there are four or five first-class installations in constant use by designers of the manufacturing firms. As instance of more important results there now exists fairly exact knowledge of the best wing sections, strut shapes, propeller blades and body resistances.

Stability and Controllability.—There are two means of obtaining stability in an aeroplane; first, by means of an automatic device, such as gyroscope, and second, by such a disposition of the surfaces of the aeroplane that the machine has inherent stability. Very little success has been obtained from the first method, but the second is now very largely employed.

Questions relating to stability and controllability are intimately connected but in one sense they are distinct. Thus it is possible to have an unstable aeroplane which is readily controllable and very popular with pilots.

Stability may be considered under three heads:—

Longitudinal stability,
Lateral stability, and
Directional stability.

The first, longitudinal stability, is obtained by means of the tail plane, and the size of this for any particular type determines within limits the degree of stability. The fundamental point, however, is the position of the center of gravity of the aeroplane relative to the main planes. If this is too far aft no tail plane can be found to give stability. The farther forward the center of gravity, the smaller is the tail plane required.

The problems of lateral and directional stability are very closely connected, and must be considered together. Lateral stability is obtained by giving the main planes a dihedral angle, and directional stability is obtained by a proper regard to the dimensions and dispositions of the fins and rudders. The area of the fin and rudder required is a function of the dihedral angle.

The outbreak of war found us in a very favorable position with regard to the development of an inherently stable aeroplane. The importance of this feature from a military point of view had been fully realized, and special efforts had been made to produce a stable and at the same time controllable aeroplane. That these efforts were successful was largely due to the late E. T. Busk, of the Royal Aircraft Factory, as it was then called, and the B.E. 2c aeroplane, which embodied the results of his work, had been fully tested and demonstrated to possess complete adherent stability prior to the outbreak of war. Unfortunately, Mr. Busk was killed in a flying accident shortly after his experiments had been brought to a successful issue. However, his full scale research had been carried so far that the principles underlying the design of this machine could be applied to any other design of aeroplane. Subsequently nearly all machines were designed for inherent stability, except such types as were considered to be more suitable for their specific work if a certain degree of stability were sacrificed for very quick maneuverability. An impression was prevalent, at any rate during the first two or three years of the war, that a stable aeroplane must necessarily be very heavy on its controls, and since quick maneuverability was an essential for fighting scouts, the aim of designers, encouraged by fighting pilots, was to obtain the maximum controllability and quickness of handling, irrespective of stability.

It was gradually realized that this was a mistaken view, and that the comparatively poor maneuverability of some of the earlier stable machines was due, not to the fact of these machines being stable, but to the particular design of the controlling surfaces. A very great amount of research into the conditions governing stability and controllability was carried out both from the theoretical and experimental standpoint, and thus revealed enough data to enable aeroplanes to be designed combining inherent stability with good maneuverability, and as a result, the prejudice against stability, in small fighting aeroplanes quickly disappeared.

A good example of the advantage of this is afforded by Captain Ball's wonderful return to our lines on his stable S.E. 5 machine after his controls had been almost completely shot away. The aeroplane practically flew itself back, and with only half of his normal elevator control he was able to make a safe landing on the aerodrome. Such a feat would have been out of the question on an unstable machine.

At the end of the war we were calling for aeroplanes with neutral stability for fighting work, *i.e.*, aeroplanes which followed the pilot's mind and hand in whatever attitude they were put. For bombing and long-distance work stability is a very important asset, as it relieves the pilot of fatigue, and facilitates the maintenance of his course and his sighting for bomb dropping. The disadvantages of not putting a sufficient amount of study into these features are demonstrated by German practice. Although a few of their small aeroplanes were fairly maneuverable, their larger ones did not compare favorably with our own when tested under the same conditions, and in the case of their very large bombing aeroplanes we had evidence that the pilot's difficulties on a long flight were very considerable, and it is probably not too much to say that the heavy proportion of German bombers which were crashed on returning from their raids was largely due to lack of stability and controllability.

Monoplane, Biplane or Triplane.—During the war all these types have been experimented with, and, in fact, have been built on a production scale, and it cannot be definitely stated that one type is more suitable than another without knowing the exact purpose for which the aeroplane is required.

Broadly, the comparative advantages of the three types are as follows: In comparison with the biplane, the monoplane is 5 per cent more efficient as a weight carrier per square foot, and can be made to afford a better fighting view. On the other hand, it is weaker for the same weight of structure, and is less maneuverable for equal total weight. Similarly the triplane, comparing it to the biplane, is 5 per cent less efficient, but is more maneuverable, and affords opportunities for a deeper and therefore stronger main girder, which gives it an advantage for the larger sizes. Incidentally, it is also more difficult and expensive to produce. The balance between advantage and disadvantage therefore depends upon the particular function required from an aeroplane, and also on the skill of the designer in overcoming the peculiar difficulties of his problem. But the general conclusion is that the monoplane is most suited for the very small aeroplane; the biplane for all general sizes, and probably the triplane for the very large sizes. Experience during the war in general bears this out, although there have been instances which might be taken as contradictory to this statement.

The importance of the successful use the Germans made of their Fokker monoplane should not be exaggerated, as the real reason for this success was the production in large numbers of a good single-seater with a rotary engine copied from the French Gnome, combined with the new synchronized guns, and a fresh method of attack. When we tested the aeroplane ourselves, and compared it with our own under similar conditions, its performance was in fact very little superior to the B.E. 2c, against which it was so effective.

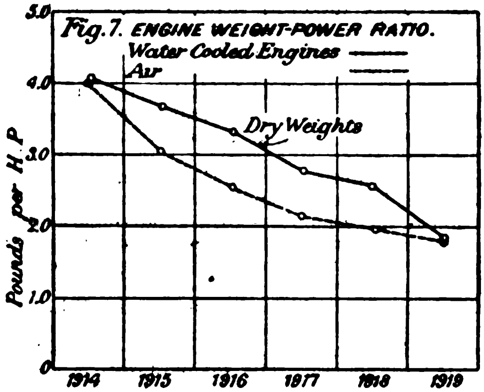
Our designers, on the other hand, made a very successful triplane fighter with a similar sort of engine. The main objectives of the designer in this case were handiness and view ahead. Handiness was obtained by very small span, and view by placing the pilot so that his eye was in the line of the center plane. However, as far as this country was concerned, the general decision was the selection of the biplane as the simplest and soundest type for all ordinary work.

Climb Requirements.—The most valuable quality in a war aeroplane and the one most difficult to supply is no doubt climb. Practically all other considerations are opposed to fast climb, such as high superficial loading, direct drive engines, high speed and long range. But the demand by pilots for a high rate of climb, both as a protection against anti-aircraft fire and to increase their fighting capacity was so great, that this quality had to be provided. The range of guns, or rather their power of accurate

shooting at a height, increased in an astonishing degree during the war. In the early days, a height of 3,000 feet was reasonably safe, and aeroplanes could do their work at that height. Then it was increased to 6,000 feet, 10,000 feet and 15,000 feet. By the end of the war shooting even at 20,000 feet was unpleasantly accurate. Whenever a new type of aeroplane appeared which could operate at an increased height, it was immune for a period, and could do its work almost unmolested, and in some cases without being perceived; but this period seldom lasted long, and sometimes even before the new type was appearing in sufficiently large numbers to count seriously, it was outranged.

The same state of affairs applied to the height at which fighting took place, which gradually increased from year to year up to 20,000 feet. In order that the pilot could fight in a satisfactory manner he required a good rate of climb as well as the uppermost position. The difficulty of providing climb in an aeroplane during the first two years of the war was enhanced by the constant increase in the number of accessories required to enable the extra military functions to be carried out. These accessories added largely to the weight and also to the air resistance, and it was not until specially designed aeroplanes appeared that the position became satisfactory.

Interaction of Engine and Aeroplane Design.—Progress in aeroplane was very much bound up with the progress in engine design; in fact, it can be broadly stated that during the whole of the war the aeroplane designers were waiting upon the engine designers, and as soon as any new engine was developed to a satisfactory point, it was but a short time before aeroplanes were in service, making the maximum use of such an engine, or at any rate a thoroughly effective use.



It is interesting to compare German practice with our own. The Germans realized at an early stage the necessity of large engines, and they concentrated their attention on the production of a first-class but simple type of motor. This had a straight row of six cylinders of an average weight per horsepower, neither excessively heavy nor particularly light. They standardized this type at a comparatively early date and consequently made early advances in production and reliability.

Our policy was by no means so definite, and our engine designers worked on a number of different types; air-cooled rotary, radial, and in V, water-cooled, eight-cylinder, twelve-cylinders in V and six cylinders in line. At no time did we rigidly standardize a single type. In consequence, we had

far bigger difficulties in production, and on the whole our motors were not so reliable. Ultimately, however, we progressed much further than the Germans, both in the power of the engine, and in the reduction of weight per horsepower.

The reduction of weight of the aeroplane per horsepower can be obtained in either of two ways: (1) By having a very powerful engine of average weight per horsepower, so powerful that the weight of the pilot and his military gear is relatively an unimportant factor; or (2) by having an engine of very light weight per horse-power. This point is best illustrated by a short analysis in weight per horsepower, comparing two aeroplanes which are both single-seated fighters, and of the same pounds per horsepower.

Large Aeroplane.	Lb./H.P.
300-horsepower engine weight	3¾
2 hours' fuel, oil and tanks	1¼
Pilot and gear	1
Aeroplane structure weight	3
	9

Total weight, 2,700 pounds.

Small Aeroplane.	Lb./H.P.
200-horsepower engine weight	2¼
Fuel, oil and tanks	1¾
Pilot and gear	2
Aeroplane structure weight	3
	9

Total weight, 1,800 pounds.

The first type has a large water-cooled engine, is economical in fuel, and by reason of the size of the engine, the pilot and his gear account for only a small proportion of the total weight; whereas on the small aeroplane the engine is a light weight air-cooled engine of heavier consumption, and the weight of pilot, etc., is a much larger proportion of the whole weight.

The two aeroplanes would practically give the same performance, but the smaller probably would be the better fighter because of its greater maneuverability. There are many other factors to take into account, such as reliability, pilot's view, ease of manufacture, etc., all of which must be considered before coming to a decision as to which is the better type.

Generally speaking the tendency during the war was towards the heavier aeroplane, although there were two periods when the light-weight engine reversed this. One was when the 80 Le Rhone Sopwith Pup supplanted the 140 R.A.F. B.E. 12, and the two 100-horsepower pushers D.H. 2 and F.E. 8; the other was when the small air-cooled Wash engine was competing with the 275 water-cooled Rolls-Royce Falcon.

Wing Structure.—General development in wing construction of aeroplanes, during the war, has been more in the nature of refinement of detail, than of evolution of methods.

With increased knowledge concerning the loads to which the structure is subjected during flight, has come the possibility of more scientific proportioning of the structural members; but beyond this the general arrangement of the wing has remained unchanged. The structure percentage weight has shown the effect of increased knowledge, and this percentage has now reached a very low value.

The effect of attention to detail has also been to give a general cleaning up and simplicity to the appearance.

In external bracing, changes have taken place in some particulars. In 1914 the duplication of the main lift wires was considered to be very important, if not essential; now, in the event of one wire getting shot away, the loads are taken through the incidence wire which are those seen when looking at a wing from the side. At the commencement of the war, hard drawn piano wire was used for the main lift and anti-flying wires in many types; when this was not used, stranded cable was employed. The use of these materials has been superseded by stream-line wires. These consist of high-tensile steel rods of a lenticular section rolled from bar. The ends are left circular, and threaded to form a simple means of attachment to the fittings. These wires have proved very satisfactory, and initial troubles, due to crystallization of the metal through vibration, having been overcome, this is now the commonest form of bracing. Hard drawn piano wire has disappeared, but cable is still extensively used, particularly in types where high performance is not essential.

The design of interplane struts has undergone certain changes. The cylinder aeroplanes had struts of stream-line section made from solid spruce, and occasionally steel tubes faired off to a stream-line form by light fairings were used. With the growth in size of aeroplanes and the increasing scarcity of suitable wood, built up or laminated struts were used. Also, with the standardization of steel tubes for aeroplane work, and an enormously increased production of these, this form of strut became very popular. It is a most efficient construction, particularly for the landing chassis.

Internal bracing wires have gone through much the same process as the main plane bracing wires—at the commencement of the war, piano wires were invariably used, but these have now almost universally been superseded by swaged rods, screwed at the ends for fitting purposes.

During the war also, the scarcity of silver spruce occasioned the development of new methods of spar construction. These were originally always in one length from the center section to the wing tip and spindled from a solid section. The first effect of timber shortage was to introduce a system of splicing, and much experimental work was carried out to determine the best form of splice to be used. The result of this work was that a straight scarf joint sloped 1 in 9 was adopted as a standard. This scarf was glued, pegged and wrapped with fabric. Its efficiency was high when compared with the solid spar, and it was a simple job when considered from a production point of view.

The splicing of spars was not found sufficient to overcome the timber shortage, and the need became evident for some change in design by which small scantling timber could be used, even for the spars of large aeroplanes. Further experimental work was put in hand, this time to determine the effect of laminating spars, *i.e.*, glueing thin strips together, and from this built-up section constructing spars in the ordinary way. At the same time experiments were made on box spars, *i.e.*, the spars, instead of being made of the common "I" section were made in box form, the flanges and webs being formed of separate pieces of materials, glued and bradded together. Both the laminated and box spars were found to be very successful, and both types were immediately put into service. No trouble has been met with from their use. Indeed, it is probable that better quality spars are obtained from this means than by cutting from the solid at any rate for large aeroplanes. The smaller the scantlings, the easier inspection becomes, and the more guarantee is there that first-class material only is being used.

Splicing and the building-up of spars have proved such successful innovations, that there is no doubt that these methods of spar construction, introduced purely as war measures to overcome the serious timber shortage, will remain as standard in the future.

Fuselage Construction.—Fuselage construction has undergone very few changes, during the period of the war, as regards methods. Larger aero-

planes have been built, and this has naturally meant greater attention to the construction, but the broad lines of design have remained fairly well the same.

There are three principal types of construction adopted in present-day practice.

1. The braced N girder fuselage (Figs. 8 and 8A).
2. The three-ply covered fuselage (Fig. 9).
3. The monocoque fuselage (Fig. 10).

The first of these consists of four longerons, or fore and aft rails, braced by a system of struts and diagonal wires in all four faces. Fabric covering completes the structure. This is the commonest type of design.

The second system retains the longerons and the vertical and horizontal struts, but in place of bracing wires and fabric, thin three-ply is glued and bradded to the framework.

The monocoque fuselage dispenses with the longerons and consists of a single sheet of three-ply or veneer molded to shape on formers.

As a general rule, three-ply frames are provided as bulkhead bracing at intervals along the length. There are several varieties of this construction, but the essence of them all is the tubular construction of thin veneer, the formers or bulkheads merely being inserted to stabilize this skin.

Of the three methods of construction, the first and third were known and adopted before the war, and modifications have been more in the nature of improvements in manufacture than in fundamental alterations in principle. The second method was, I believe, first used by the Germans in their early Albatross scout at the beginning of the war, but has since been used with great success by British designers, notably in the case of the De Havilland designs.

Monocoque Construction.—The monocoque method of construction was adopted by the Germans at an early date and was later almost universally used by them. The probable reason for this was their ample supply of high-grade three-ply, rather than any constructional or aerodynamical advantages which they attached to this method of construction.

The chief advantage from the service point of view, of a monocoque method of building fuselages lies in its invulnerability to casual bullets. In an ordinary fuselage a bullet striking a main longeron or strut might quite possibly cause the structure to collapse, whereas a monocoque will stand up to any number of bullets. A disadvantage is that truing up becomes difficult, and special arrangements are necessary to overcome this defect.

Our supplies of three-ply were deficient almost up to the end of the war, owing to the failure of the Russian supplies, and this no doubt had a considerable influence on our designers and manufacturers. One of our firms did develop a very simple method of constructing in three-ply, which would have admitted of a large production, but the particular aeroplane was not adopted for other reasons.

It was thought at one time that the rounded fish-like form of monocoque would considerably lessen air resistance, and would enable a higher speed to be developed. The importance of this, however, has been a good deal exaggerated; the advantage gained is not worth very much.

The monocoque construction offers great opportunities of development for commercial aeroplanes, as it leaves the space inside the skin free from obstruction, a very considerable advantage when carrying mails, goods or passengers, and in this respect it has a great pull over the standard braced fuselage where diagonal bulkhead bracing wires occur at every panel. It has also a considerable advantage in the almost complete absence of metal fittings, making for ease of construction.

The largest aeroplane yet built—the Tarrant Tabor—has a fuselage 11 feet in diameter, which has been built on a modification of this principle, the skin being made of layers of thin laths glued and tacked over formers

until set. The skin is stabilized by latticed hoops of wood, and the resulting structure is a very light and strong fuselage with the whole of its interior free from obstruction.

The regrettable accident to this machine on its first trial in no way reflects on the basis design of the machine.

In a flying boat, the hull has to function both as a boat, as a landing carriage, and as a fuselage, and it is not uncommon to hear of the seaplane, as a class, being under serious disadvantage to the aeroplane because of the heaviness of its hull. Our experience during the war has brought us to the point of being able to state that a well-designed hull is no heavier than a fuselage and landing carriage, and that sea aircraft are therefore under no disadvntage under this heading as compared with land aircraft.

Metal Construction.—There is no doubt that the future of metal construction for aircraft is very promising. The amount of work and time entailed in designing metal structures is very much greater than that required for wooden ones, owing to the experimental work involved. There is also the difficulty of introducing modifications during the production stage, modifications which were continuous, during the war, owing to service demands to meet new and unexpected conditions. During the war, therefore, metal construction has not been generally adopted. Its future, however, was clearly seen, and a great deal of experimental work was done, principally in connection with the Avro Training aeroplane. At the end of 1917, as I have said before, there was a shortage of good quality timber, and a still greater shortage was foreseen as probable in the future, owing to the demands of the enlarged programme. A big effort was made to employ metal for the Avro which was absorbing something like one-third of our total wood supplies, and which was a thoroughly well-established type, with an assured future for training purposes.

The experimental work done in this direction was very considerable, and much interest was aroused among manufacturers. As a result, it can be definitely stated that, even for such a small type, the use of metal enables the wings to be made slightly lighter and distinctly stronger than is possible with the best quality spruce. The experimental work necessary to get at this result was so extensive and protracted that the type was not actually produced in metal in quantities, but the information obtained was of the utmost value and was well worth the effort.

Experimental work was also carried out in the application of metal construction to larger aeroplanes, and there is no doubt that the constructional gain both in weight and in reliability will be more marked in the case of large types than in small ones. The uniformity of strength obtainable in metal will give it a marked advantage over wood, and will enable aeroplanes to be made without such large allowances for variations of material.

The Germans, on the other hand, were driven to the necessity of employing metal at an earlier date than ourselves, and although they never turned out any metal structure which would have satisfied us, this form of construction was very considerably employed. Their A. E. G. twin engine bomber was almost entirely made of metal, but it was relatively heavy and allowed only a very small weight of bombs or fuel, before it became dangerously overloaded. For the spars they used high-tensile steel tubes which are not altogether satisfactory for use as a combined strut and beam.

At the end of the war there certainly did appear an all-metal German aeroplane which was called the Junker monoplane. This was of a quite novel type of construction, the wings being covered with corrugated aluminium sheeting and having multitudinous internal tubular bracing. A very thick wing section was employed. The performance was not good and it is doubtful if the type would have really been of any serious value, but as a piece of construction it showed considerable merit.

The Progress of Design and Construction of Propellers.—The screw

propeller has for many years been used as an organ of propulsion for marine craft. It was not, therefore, a totally new problem with the advent of aircraft. In its use on aircraft the propeller altered its shape, and the blades became longer and narrower as compared with marine practice.

At the outset trial and error methods were applied, and a suitable propeller was arrived at only after the trial of a large number of designs. The variations met with in aircraft work are very great. At the present day, aircraft speeds vary from 40 miles per hour in the airship to 150 miles per hour in the aeroplane. The engines vary from 40 horsepower to 600 horsepower, while the propeller revolutions vary from 500 per minute to 2,200 per minute. With such wide variations in conditions a surer and quicker method of arriving at the best results was demanded. Curiously enough, aircraft propeller designers had recourse to a method devised by M. Drezweichi in 1882 for the design of marine propellers, which consisted of a mathematical analysis treating the propeller blade as a rotating wing. The method was tested at the National Physical Laboratory, and is now in general use, with the result that it is rarely necessary to test more than two propellers to arrive at the desired result. Frequently the result is obtained with sufficient accuracy in the first trial design.

The efficiencies now obtained in aircraft propellers are high, being 75 per cent as an average, and frequently 85 per cent under optimum conditions.

As regards construction, the peculiar vibration and variation of torque prevented the use of metal, which quickly fatigued and failed. For this reason and also for quickness in manufacture, wood became and still is the principal material used.

Only the highest class material was suitable, for, at the speed at which modern aircraft engines run, the radial pull at the root of a propeller blade may be as much as 5 tons, while at the same time the propeller has to pull the aircraft along, involving bending moment due to forces of as much as ½ ton per blade. The stresses due to these combined forces were satisfactorily met with for the smaller powers, but at one time it was thought that the size of engines would be limited by the maximum power which could be transmitted through a single propeller. And such might have been the case, had it not been discovered that instead of the stresses due to these two forces adding to each other, it might be arranged so that the bending stresses due to centrifugal action would completely neutralize those due to the thrust.

The only limits now imposed on propeller design are:—

1. Tip speed.
2. Constructional limitations.

As regards the first, experiments have shown that a tip speed equal to the speed of sound should not be exceeded, and to this speed we are rapidly approaching. In the standard propeller for the American Liberty engine in the D.H. 9A, the tip speed is 880 feet per second as compared with 1,050 feet per second for sound. This difficulty, however, is met by gearing the engine so as to reduce the propeller speed. Provided the engine is suitably geared so that the tip speed does not exceed 900 feet per second, there appears to be practically no limit to the amount of power which can be transmitted through a single propeller. Even with present-day methods of construction it is safe to say that 2,000 horsepower can be transmitted through a propeller mounted on an aircraft flying at 80 miles per hour and as much as 6,000 horsepower on an aircraft flying at 150 miles per hour.

As regards the question of commercial construction, development has been along normal lines; improvements have been made in the details of manufacture such as the glueing of the numerous wood laminations, and the protection of the blade with fabric and metal edgings; and the general

result has been a better and more reliable article. Nevertheless, the use of wood is not wholly satisfactory, especially for future commercial work, involving flight through tropical countries, and we therefore look forward to the solution of the problem of the metal construction of propellers.

Division into Types.—When the war began the only aeroplane function which was considered seriously by the army authorities was that of reconnaissance, and there is no doubt that the B.E. 2A, which was developed by the Royal Aircraft Factory for this function was an excellent aeroplane, thoroughly well adapted for its work.

Reconnaissance Aeroplanes.—These early aeroplanes were not intended to carry any military load other than crew, but almost from the very start of the war the military load required began to increase, with the result that all through 1915 and well into 1916 our reconnaissance aeroplanes were overloaded with all kinds of equipment, such as guns, gun mountings, cameras, wireless apparatus, etc., which were attached to any available part internally or externally. In order to carry out their requisite functions, this additional equipment was necessary, but it naturally meant a very considerable reduction in performance. The speed was brought down 10 miles or 15 miles an hour, while the extra weight and resistance had a very serious effect on the rate of climb. This is shown by the diagrams giving the performances at the different periods of the war; it is not until the latter part of 1916 that any appreciable improvement in performance can be seen, and then both the speed and the climb go up rapidly.

It was at this period that the special military aeroplanes, designed subsequently to August, 1914, first came into use, and in these full allowance was made for all equipment to be carried. The increase in performance was due not only to the higher power engines used, but also to the fact that the aeroplanes were designed to carry the equipment as far as possible internally, thus avoiding all unnecessary resistance.

As aerial tactics developed, the roles of the pilot and observer altered, and the latter practically confined himself to being a pair of eyes in the back of the pilot's head, whereas the pilot had to do all the reconnaissance artillery, spotting, etc., as well as fly the aeroplane.

Single-Seater Fighter Development.—As well as doing one's own scouting, it was necessary to stop the other side doing his or rather to stop him from taking his reports home. This meant fighting, and fighting meant guns and better performance. The early aeroplanes were soon fitted up with a Lewis gun fixed so as to fire over or at the side of the propeller disc, and actuated by a Bowden wire. This gun was at first fixed to fire upwards at a considerable angle to the line of flight, necessitating an oblique method of attack. Then later, in order to enable the aeroplane itself to be aimed directly at the objective, the gun was mounted on stands and in a line practically parallel to the propeller shaft. It was found that, by aiming the aeroplane itself rather than an independent gun, shooting was more accurate. The attachment of all this gear and weight to machines which were not designed for them was a very serious handicap, and performances were reduced 15 per cent or 20 per cent on speed and ceiling.

Late in 1915 the Germans brought out a system of synchronizing the trigger with the engine, so that the bullets went between the blades of the propeller. This was a great improvement as it enabled the gun or guns to be fixed alongside the engine, within easy reach of the pilot, and offering very much less wind resistance. The Vickers gun was quickly adapted to work in a similar manner on our tractors, and by the late summer of 1916 our single-seater fighters were on a par with the Germans'. During this period, in addition to the tractors with exposed Lewis guns, we had been making very considerable use of small single-seated pushers of remarkably good performance considering the difficulties of the proposition.

The characteristics of the Sopwith Pup, our first good tractor single-seater, were very light surface loading, a small but good rotary 80-horse-

power French engine, and every scrap of unnecessary weight eliminated by careful design. The view, particularly overhead, was not very good, but the aeroplane was so handy fore and aft that this did not interfere very seriously with its fighting qualities. This type lasted a very considerable time before it was superseded, which, in view of the comparatively small horsepower, was remarkable. During the period in which this type was in use fighting acrobatics advanced to a marked degree, and in the next type an effort was made to increase view and maneuverability. This was the Sopwith triplane with another French engine of 110 horsepower to 130 horsepower.

The loading per square foot had increased, but the loading per horsepower had decreased considerably, and the performance was better. Some of the aerodynamic disabilities of the triplane were overcome by the pronounced forward stagger and the use of a single strut. This single strut system increased the difficulties of manufacture and repair, particularly as regards truing up. Both this aeroplane and the one before it were provided with only one synchronizing gun, and the rate of fire was consequently slow.

In aerial fighting, the time in which it is possible to hold the enemy on the sights is very short, and one gun was found insufficient, apart from the considerable chance that the one gun might jam at the crucial moment.

In the next type, which was in design at the end of 1916, and came into production in 1917, we see the influence of these considerations upon the mind of the designer, who by the way was Mr. Hawker, of transatlantic fame. We have here a very simple and, though unstable, an easily controllable biplane with two guns. Enormous numbers of this type were produced in the course of the next year or two with French and English engines which increased in power up to 150 horsepower. The type was nicknamed the "Camel" because of the curious hump in the fuselage.

The next type on our list is the S.E. 5 designed at the Royal Aircraft Establishment, Farnborough. This is really a direct descendant of the B.E. 2A with which we started. It was a very long time coming into general use, principally owing to delays with the engine, a 200-horsepower water-cooled French Hispano. Opinion was by no means unanimous as to the fighting value of this aeroplane compared with the "Camel," and each type had its school of adherents. The S.E. 5 had a better performance, particularly at a height, was stable and maneuverable, very easy to fly, and had better visibility due to the pilot being farther back. On the other hand, it was much more difficult to produce and the engine and its accessories gave a great deal of trouble both at home and in the field. By this time we had reached a point where the performance of our aeroplanes was very considerably in advance of that of the Germans, who were cramped by their non-elastic engine policy.

The next type to come into general service was the Sopwith "Snipe." The engine here was a 200-horsepower B.R. 2 of British design. It would appear that this size of air-cooled rotary engine has reached, if it has not in fact surpassed, the maximum size for efficiency. The performance was rather disappointing and must be attributed to the enormous engine diameter and air resistance. Several competitive aeroplanes from different makers were tested simultaneously with this one, and the results were practically all the same. The control surfaces are balanced. In this way the necessary maneuverability was maintained, although the weight of the aeroplane had reached rather a high figure. The actual selection of this type was largely governed by the engine position which entailed the placing of orders on a production scale before trials of the actual engine and aeroplane were carried out.

Finally we come to the Martinsyde F. 4 which, although in production, was never used at the front previous to the signing of the armistice. This had another water-cooled engine—275-horsepower Rolls-Royce of high efficiency. Both aeroplane and engine were very good, and although the

former was heavy and somewhat large, it was wonderfully maneuverable. The performance reached was very high—over 130 miles an hour at 15,000 feet with a climb to that height in 12 minutes.

The increase in speed and climb from period to period is shown on Figure on page 769.

Two-Seated Fighters.—Apart from the particularly interesting case of the single-seaters, there developed from the use of the original reconnaissance type, a demand for, and a supply of two-seated fighting aeroplanes, which could do their reconnaissance work and defend themselves, if attacked. As the best method of defence has always been, and will be, to attack, these machines which were in principle defensive, became very effective in the offensive. Our two-seated pusher fighters, such as the Vickers or the F.E. 2s, gave a very good account of themselves, until the time came when their poor performance, as compared with the tractors, put them out of date.

The first really good two-seated tractor was the 1½ Strutter, so called by the Sopwith Company because of its peculiar wing strut system. This aeroplane was, and still is, one of the most efficient ever designed, and for its engine power (130-horsepower Clerget) it has never been surpassed. In it the pilot was placed as close up against the engine as possible with a very fine view ahead and downwards. The gunner was now put behind him with a rotatable Lewis gun turret. He had a very good view downwards and all round the rear. This arrangement, which is, in principle, still standard, gave a very maneuverable aeroplane owing to the concentration of the weights; and on account of its high performance the type became a very effective offensive weapon.

Early in 1917 there appeared another two-seated fighter where the qualities enumerated above were still further developed. This was Captain Barnwell's Bristol Fighter with a Rolls-Royce Falcon engine. Both aeroplane and engine were exceedingly good. The view from the pilot was improved by putting him farther back and higher so that he could see over the top plane, and being closer to his gunner, they could communicate better. Close communication in this sort of fighter is a very vital matter, and probably this type did more to establish our superiority over the enemy than any other.

Bombers.—Reverting to our original B.E. 2As., from which we have seen the development into one-seated fighters and two-seated bombers, we can trace also the development of the bomber. This was the type of aeroplane first used for bombing purposes, as soon as bombing, as a useful and military function, was appreciated. There were sporadic efforts at bomb dropping by 80 Gnome Avros, but when it developed into organized attacks with numbers of aeroplanes, the original reconnaissance type was used. As soon as the real requirements of this work were understood specially designed aeroplanes appeared. In the early types the bombs were fitted in all kinds of ingenious but crude ways, with pieces of string and cutting knives, etc., and the attacks were made by diving to a low height and planting the bombs by the pilot's judgment. Although great efforts were made in the training of bomb droppers and in the perfection of sighting and releasing apparatus, etc., accuracy from a height was never really obtained, and experiments conducted towards the end of last year, proved conclusively that the only way to hit any isolated object with certainty was to dive to a low height before letting go the bombs. Nevertheless the moral effect of bombing is immense in disturbing industrial work and the manufacture of munitions.

With the introduction of the B.E. 2c, systematic bombing of the enemy's positions was carried out, but this aeroplane was capable of making comparatively short raids only, and when loaded with bombs could not carry an observer. With its moderate speed it was rather an easy prey for enemy scouts under such circumstances.

The Martinsyde single-seater bomber with the 120 and later the 160 Beardmore engine was a distinct advance as regards range, while its greater speed made it a much better fighter.

As bombing developed, a subdivision of type into definite branches took place. These were day bombers, short distance and long distance; and night bombers, short and long distance. From an aerodynamic point of view the design of a high-speed, long range weight-carrying aeroplane is an exceedingly difficult problem, but it can be fairly stated that our designers, particularly Captain De Havilland of the Aircraft Manufacturing Company, found the solution in advance of those of other countries. His aeroplanes eventually had wing surfaces of high efficiency and of high superficial loading, combined with structures of great cleanliness and low air resistance, and with the most powerful engines available at the time. Owing to their low weight per horsepower his machines could get off easily though at a high speed, and were very fast in the air. On returning home, after the fuel was consumed and the bombs dropped, they could land easily enough, but if by chance they had to land with full load on board, the pilot's difficulties were considerable.

Just as the original B.E. 2 was the first used for bombing by day, similarly it was the first type to be largely employed for bombing by night, for which function it was very fairly suited, up to the limit of its capacity, as it was very easy to fly and to land. The weight-carrying capacity was, however, not great, as the engine was only 70 horsepower to 100 horsepower. When the bigger engines became available this bombing work was taken over by the F.E. 2 pushers, which were good weight carriers and had a better view for bomb sighting.

The above remarks apply more to the military flying corps than to the naval service, which made use of any aeroplanes they could get hold of, particularly French ones, such as Caudrons and Farmans. Later, they used the Sopwith 1½ strutters, and finally Handley-Pages. This latter type was the first really effective design of night bomber, and came into use early in 1917. Its general features are so well known that it is unnecessary for me to enlarge upon them. When, in the autumn of 1917, a big program of bombers was laid down, this was the type decided upon. The Vickers-Vimy was a later machine of the same general type embodying more recent experience.

At the end of 1917 it was considered to be worth an attempt to make a very long range bomber, to attack Berlin itself, from a base in England. The actual point-to-point distance is about 450 miles, and a minimum range of 1,100 miles was decided upon. As the aeroplane would have to fly by day, as well as during the night, it was essential to provide a very complete gun defence system. This meant a crew of seven, with many guns and much ammunition, apart from wireless telegraph apparatus and bombs. It was clear that it would be necessary to employ a very large aeroplane with powerful motors. The 350 Eagle Rolls being the biggest and best geared engine available, it was decided to employ four, using a system of tandem propellers, which theoretical investigation showed to be reasonably efficient. The main lines of the design were quickly arrived at, and arrangements made for the building of a few experimental aeroplanes, by a large firm of shipbuilders in Ireland. In order to get a straight run on the design and to avoid interruption, Mr. Handley-Page, the designer, went over to Ireland with a large proportion of his design staff, and after six months of very hard work, the first of the batch came over to Cricklewood, to be erected and tested. Many difficulties, particularly connected with the control, were encountered and surmounted, and on November 8, three days before the armistice was signed, two of these aeroplanes were standing ready and fully equipped to start for Berlin.

GROUND FIGHTERS.

Armoured Aeroplanes.—In the very early days of the war the aeroplane had little to fear from enemy action provided it crossed the lines at any height above about 3,000 feet, so as to be out of range of ordinary rifle fire from the ground. The introduction of the special anti-aircraft gun, however, very quickly altered this state of affairs, and the aeroplane was driven higher as the accuracy of fire improved; until, at the end of the war, there was no immunity from danger from anti-aircraft fire at a height of 20,000 feet.

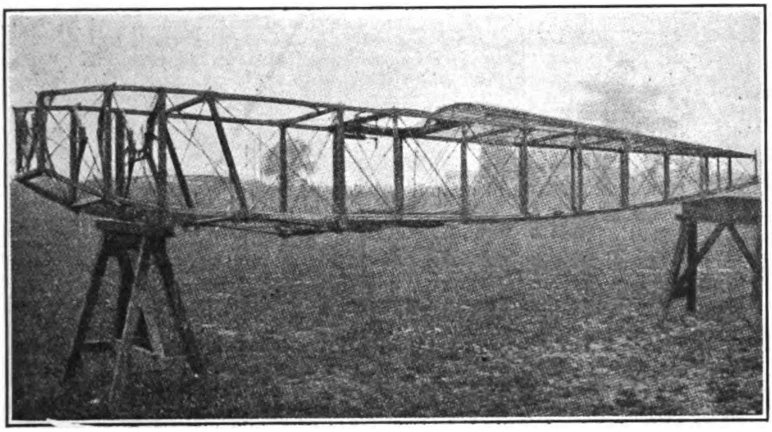
For a long time it was considered that the risk of a low-flying aeroplane being hit by rifle or machine-gun fire from the ground was so great that we should not be justified in using our pilots and aeroplanes in this way. The remarkable escapes experienced from time to time by pilots forced to return across the lines at very low altitudes, however, encouraged the idea that the risks of low flying had been a good deal exaggerated, and a most effective mode of attacking the enemy was gradually developed. The first important success in this direction was, I believe, achieved by Lieutenant-Colonel Bishop, V. C., who attacked an enemy aerodrome soon after daybreak. After riddling the officers' and men's quarters with bullets from his machine guns he turned his attention to the German aeroplanes which had been brought out to attack him. He put two or three out of action on the ground, brought down three in succession as they got into the air, and then returned safely to his own aerodrome. Thereafter this method of attack developed very rapidly, and it proved of the greatest value later on, not only in breaking up the dense infantry formations advancing to attack us after March 21, 1918, but in demoralizing the retreating enemy when we subsequently advanced. A system of co-operation between our low-flying aeroplanes and the infantry was also evolved, and this proved of the greatest value in keeping the infantry and the headquarters behind in touch with what was going on in front. The aeroplanes used were the single-seater fighter—especially the light, maneuverable "Camel"—for attacking ground targets from very low altitudes; and the standard artillery machine for the infantry co-operation from rather greater altitudes.

Our casualties in these operations were naturally heavy, and the problem of providing a suitable armored aeroplane arose. As a result the Sopwith "Salamander" was designed, and was just about to make its appearance in the field at the time of the armistice. This was a single-seater fighter with B.R. 2 engine, equipped with two Vickers' guns and a very large amount of ammunition for attacking ground targets, while it carried nearly 650 pounds of armor plates, so arranged as to give the pilot and all vital parts of the aeroplane adequate protection from German armor-piercing bullets fired from the shortest ranges. In spite of this heavy load it had a speed of 125 miles per hour, and was sufficiently maneuverable for the pilot to make the attack by diving at the target, thus getting the advantage of firing in the direction of flight. Had the war lasted a few months longer this aeroplane would probably have done most effective work in hastening the German retreat.

An armored two-seater for infantry co-operation was also designed and produced, but like the "Salamander" it was too late to come into actual service.

PART III.—PROGRESS IN ENGINE DESIGN.

Before 1914 it may be said that aero engine design had merely reached the stage of sufficient reduction of weight/power ratio to enable aeroplanes to fly. Beyond this, practically no special adaptation to aeroplane requirements had been attempted. The development of the aerial arm in the war, however, speedily emphasized the importance of other factors. The



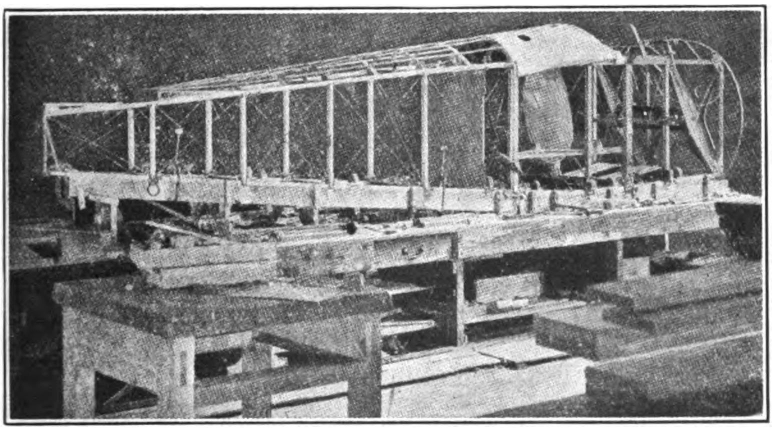
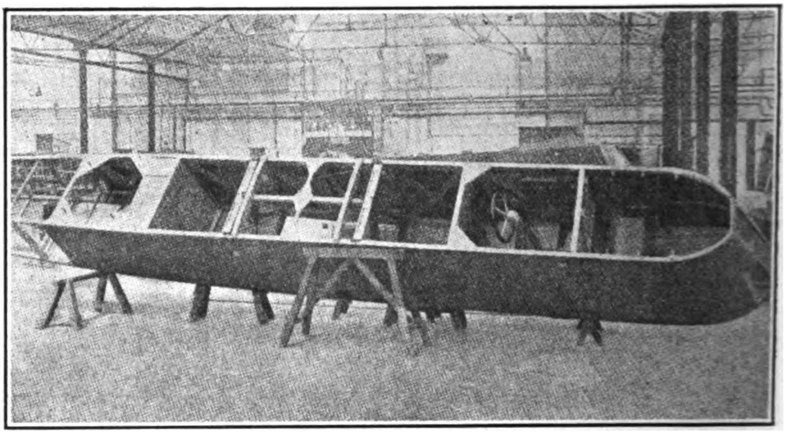


FIG. 8.—BRACED N. GIRDER FUSELAGE. FAIREY.
FIG. 8A.—BRACED N. GIRDER FUSELAGE. CAMEL.



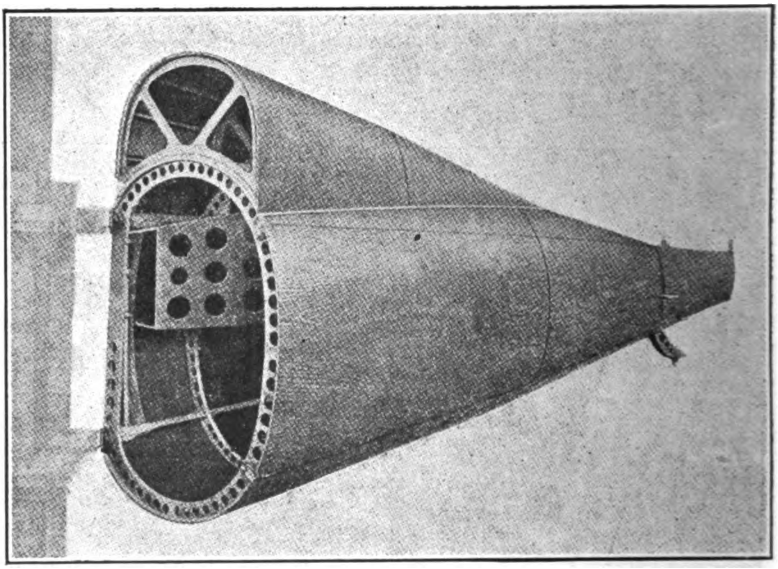
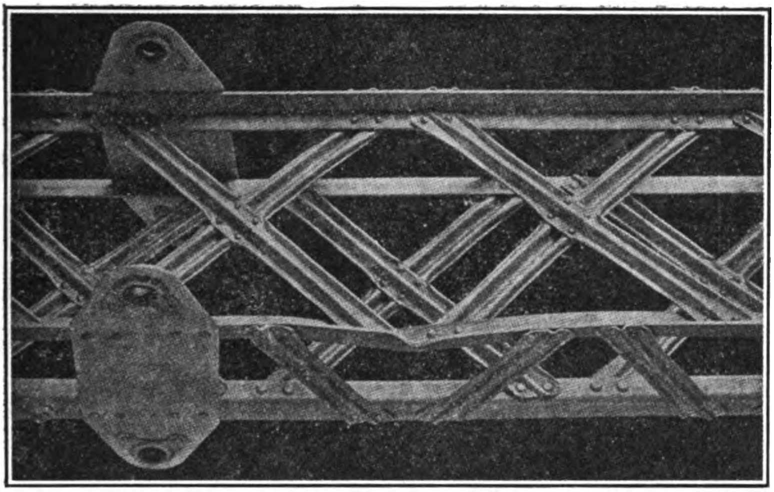


FIG. 9.—THREE-PLY COVERED FUSELAGE. DE HAVILLAND 10.
FIG. 10.—TYPICAL MONOCOQUE FUSELAGE (PARNALL PANTHER).



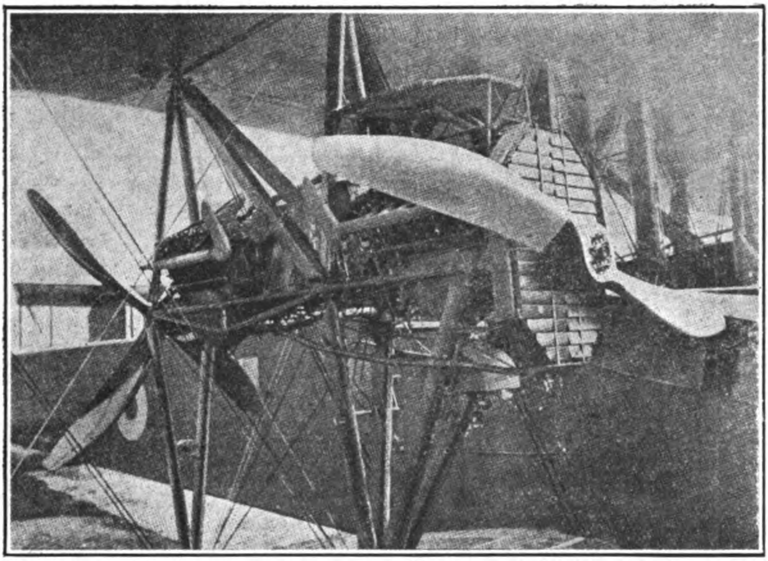
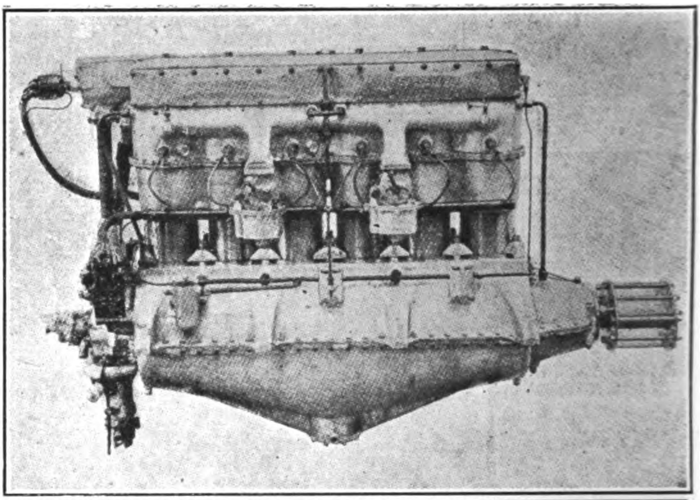


FIG. 11.—METAL CONSTRUCTION. VICKERS DURALUMIN.
FIG. 12.—TANDEM PROPELLERS. HANDLEY PAGE V. 1500.



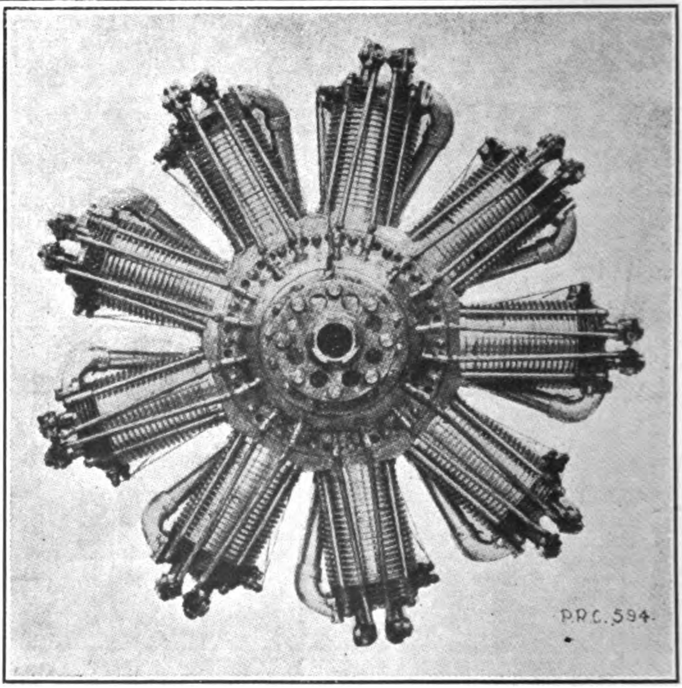
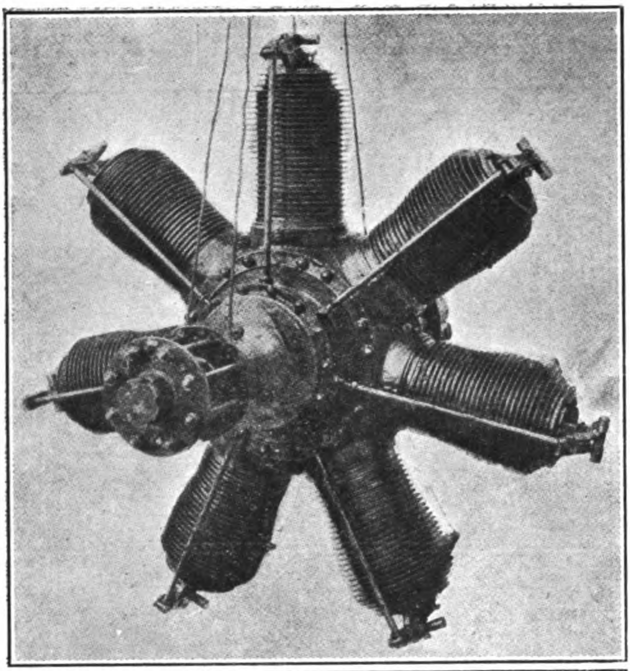


FIG. 14.—230 "PUMA" (SIDDELEY).
FIG. 15.—BENTLEY ROTARY.



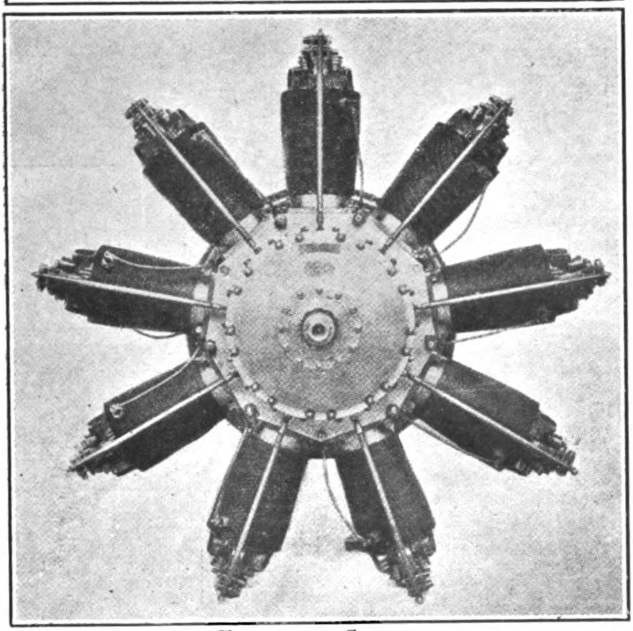
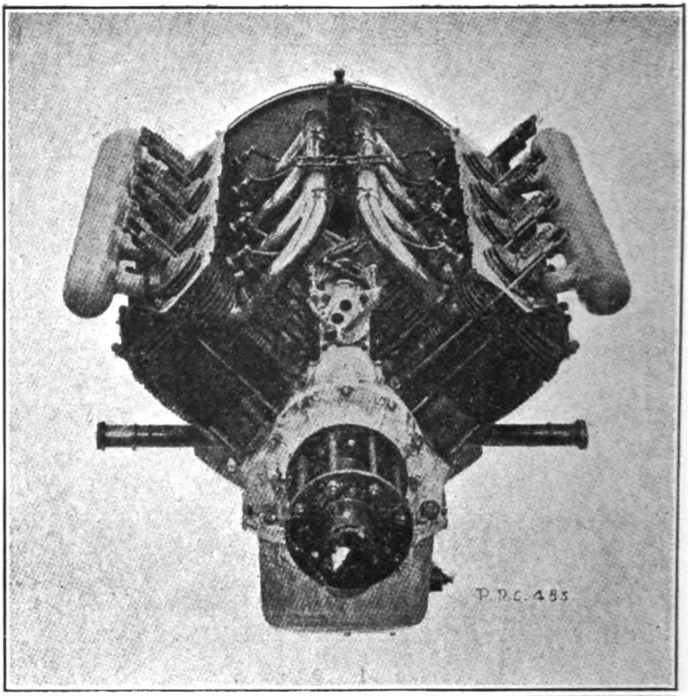


Fig. 16.—80 Gnome.
Fig. 17.—A.B.C. Dragonfly.



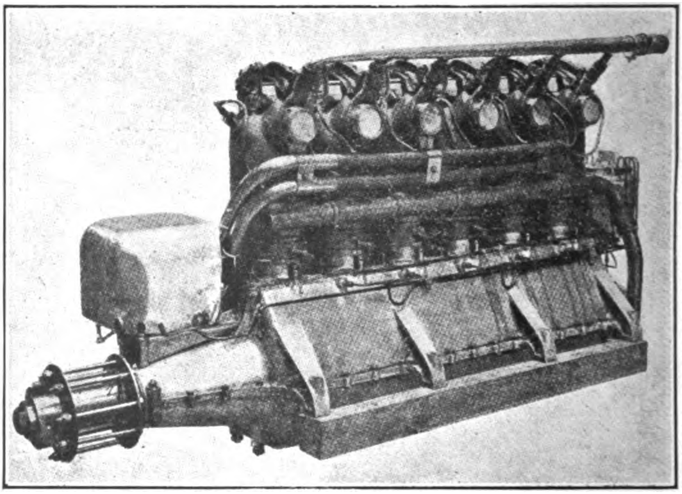
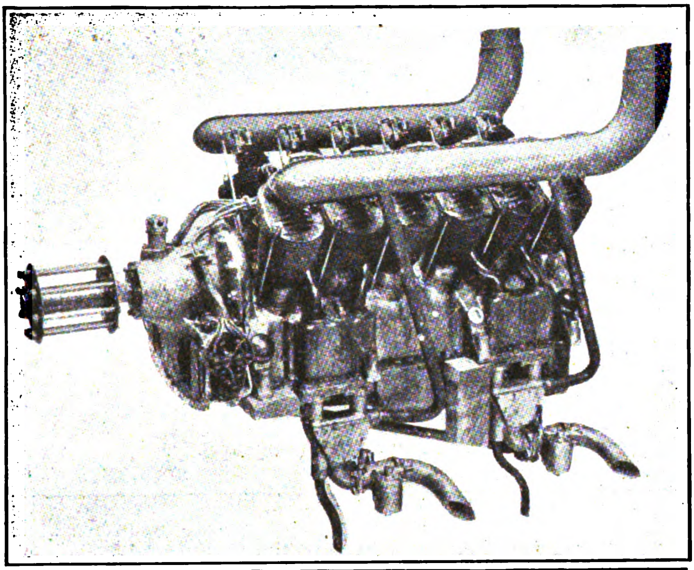


FIG. 18.—RENAULT.
FIG. 19.—160 BEARDMORE.



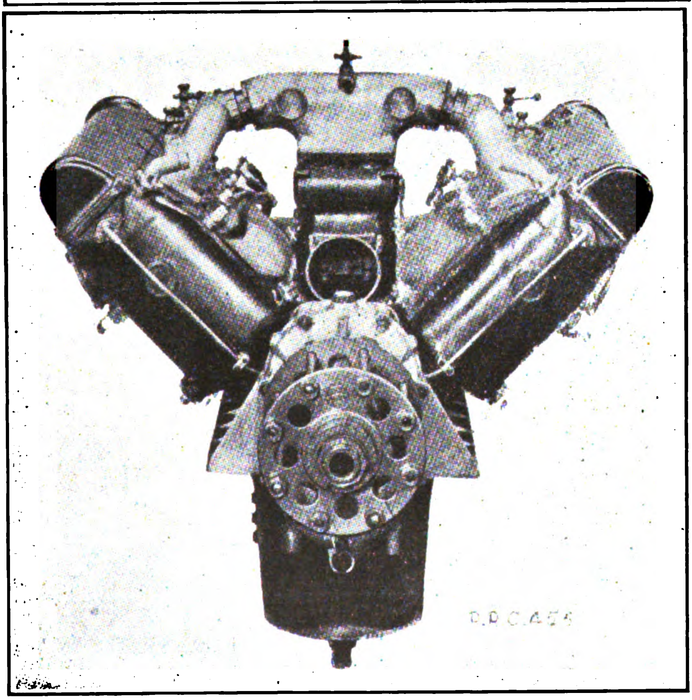


FIG. 20.—140 R.A.F. 4A.
FIG. 21.—HISPANO-SUIZA.

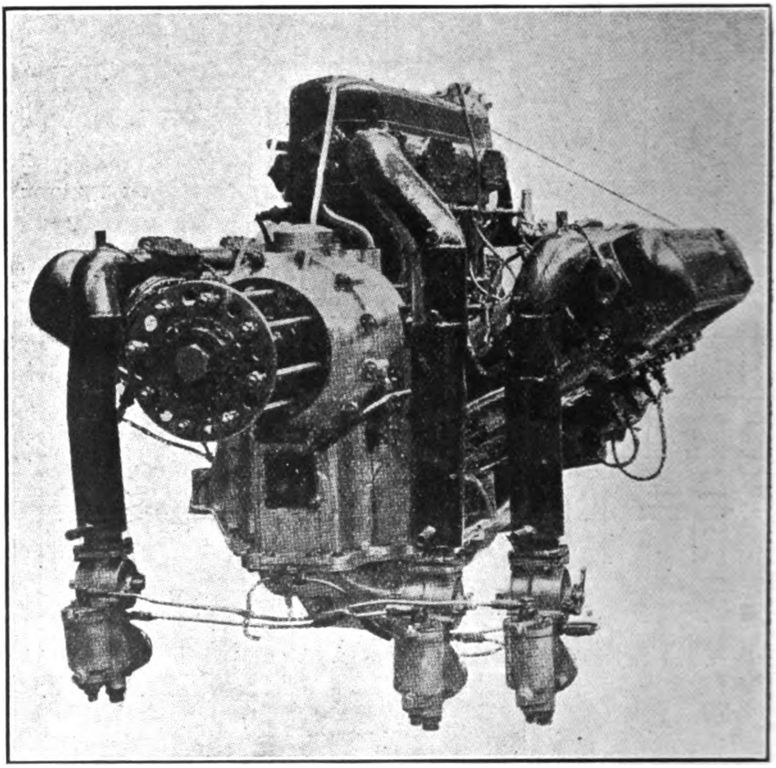
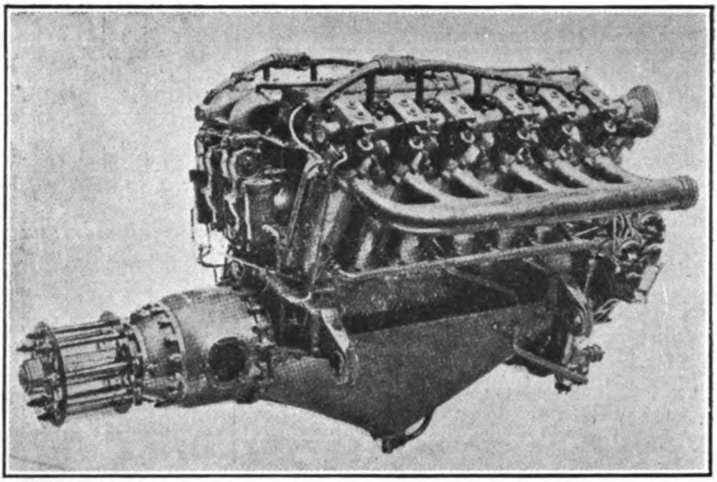


FIG. 22.—NAPIER LION.



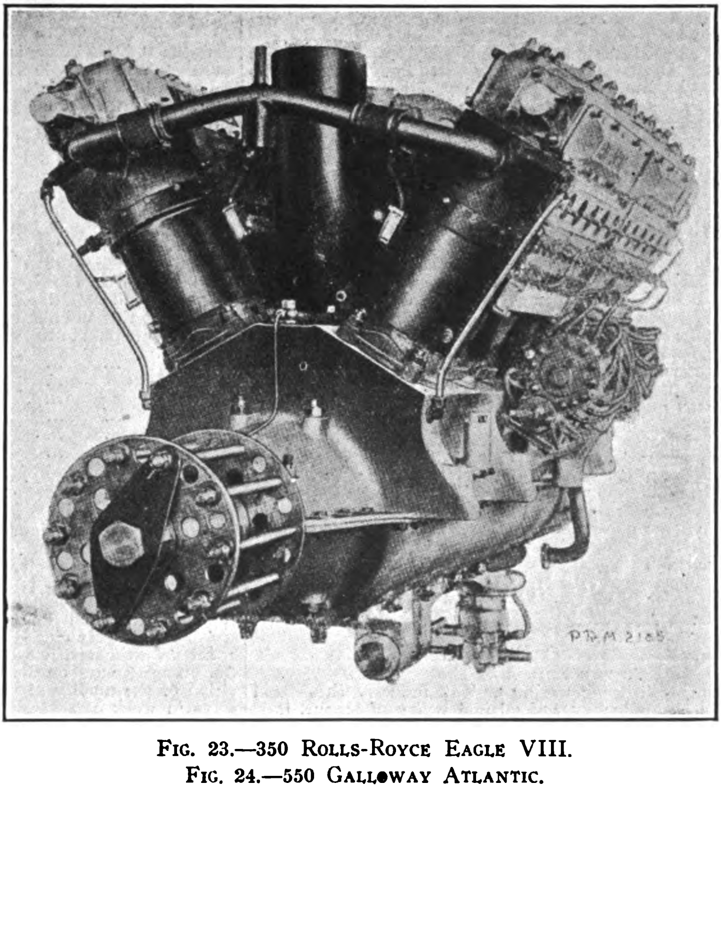


FIG. 23.—350 ROLLS-ROYCE EAGLE VIII.
FIG. 24.—550 GALLOWAY ATLANTIC.

extension of the range of operations and the losses of machines, due to engine failure, raised insistent demands for greater reliability, whilst the necessity for easier maintenance called for engines of greater all-round accessibility.

The rapid progress of flying skill and the adoption of aerial acrobatics in fighting created a demand for engines of short length and quick controllability.

The desirability of carrying out reconnaissance untroubled by enemy attack or anti-aircraft fire focussed attention on the necessity of increasing the ceiling or maximum height of operation. Coping with this most important requirement again produced an improvement of the weight/power ratio, and the ceiling of approximately 7,000 feet in 1914 was increased to nearly 30,000 feet by the beginning of 1919. It must be remembered also that the diminished density at great height decreases the amount of oxygen taken into the engine, and therefore the power which the engine can deliver. At 15,000 feet, for instance, a 300-horsepower engine can only deliver about 200 horsepower.

The extension of bombing activities at long ranges emphasized the importance of the study of the fuel consumption which, for a flight of 6 hours, amounts to a large weight per horsepower. The question of the thermal efficiency had therefore to be studied in conjunction with the weight-power ratio, so that engine speeds increased whilst the aerodynamic requirements of the machine demanded reduced propeller speeds.

Long before the end of the war the power requirements of some aeroplanes and seaplanes had far outstripped the possibilities of any one engine, so that machines possessing two, three, four, or even more engines, were in service or being built.

An important factor affecting aero engine development is the time which is required to produce a new design. Generally about eighteen months would have to elapse between the commencement of the design and a useful flow of reliable production engines. During most of this period no useful practical experience can be obtained as to the qualities or defects of the new design, and by the time bulk experience of its behavior in service is available, it is necessary to supersede it by another more advanced type. Less than half the time is required for the development and trial of an aeroplane design, so that the aeroplane is generally well ahead of the engine for which it is designed.

In 1914 our aircraft engine position was by no means satisfactory, and we depended for a large proportion of our supplies on other countries, principally on France, whose Gnome and Renault engines were pre-eminent. Great efforts were immediately made to extend our sources of design and supply, and by the end of the war British engines had gained foremost place in design, and were well up to requirements as regards supply.

Many of the earlier designs were considerably influenced by previous automobile engine practice, but a wide divergence of design and detail soon took place due to the entirely different nature of the conditions to be faced. In automobile practice silence and good carburation over a wide variation of speeds and loads were the most important features, whereas those points are of small importance for an aircraft engine. The task of the aero-engine designer was still further complicated by the fact that the order of importance of the various features of the engine is different according to the class of machine for which the engine is being designed. Certain entirely novel conditions had to be met and their attendant difficulties overcome. For example, an aero engine must have the ability to function in practically any position and, for a time at least, when completely inverted. This requirement has had a far-reaching effect on the lubrication system of aero engines, as it practically precludes the carrying of oil in the crank-case.

The shape of the engine is a matter requiring careful consideration for aircraft, as head-resistance, accessibility and small moment of inertia are

all features of considerable importance dependant on the shape. The wide ranges of temperature and pressure through which an aeroplane may pass affect the carburation, the cooling system, the lubrication system, and even the ignition, to a very serious degree. An aeroplane may undergo a very rapid change of as much as 75 degrees F. in temperature, combined with the maximum difference in moisture content of the air. With water-cooled engines, therefore, it has been found necessary to put a thermometer in the circuit and fit the radiator with blinds operated by the pilot, and even with such accessories the maintenance of the water at a constant temperature has often been a matter of great difficulty, and has thrown a heavy responsibility upon the pilot. At the same time evaporation losses must be reduced to a minimum, as the amount of water lost on long journeys is an important feature, so much so that it is true to say that the Atlantic could not be flown today with any of the water-cooling systems that were deemed sufficient at the outbreak of war.

The intense vibration due to the conditions of high speed and lack of rigid support under which aero engines must work impose new and severe conditions mechanically on every part. The violent and varying slip stream from the propeller also imposes a new problem as regards carburettor intake conditions, where again the seriousness of fire risk has to be taken into account and avoided. Probably no detail of the whole engine process has received more expert and prolonged attention than the ignition, and much of the increased reliability and efficiency of the modern engine undoubtedly results from this work.

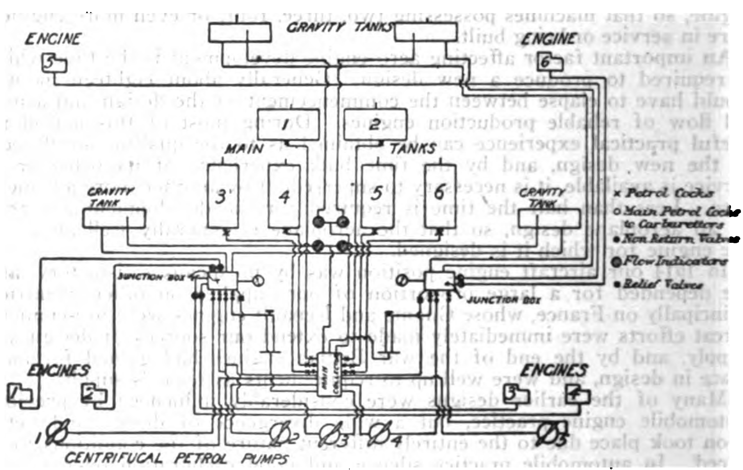


FIG. 13.—PETROL SYSTEM.

The very severe centrifugal and inertia effects which are experienced in aerial fighting, coupled with the necessity under such conditions of immediate response to the throttle, have necessitated careful design of the petrol supply system to ensure a constant and adequate supply of fuel.

In addition, the length and ramifications of the fuel system have increased considerably with the growth in the size of machines and in the number of engines. Fig. 13 shows a typical lay-out of petrol system for a large modern multi-engined machine. The main tanks are numbered 1, 2, 3, 4, 5 and 6, and have a combined capacity of 1,800 gallons of petrol.

These six tanks feed by gravity to the main collector which is situated in the engineer's cabin. In each of the pipes between the collector and tanks is fitted a non-return valve to obviate any possibility of the petrol running from the top tanks into the bottom as these six tanks are not all on the same level.

From this collecting box the petrol is lifted by five centrifugal pumps, which are shown below, into the junction boxes, which feed the engines. These centrifugal petrol pumps are driven by airscrews which are placed in the slip stream of the propellers in order that the petrol shall be pumped up whilst the machine is stationary. If these pumps should fail to act the petrol can still be lifted to the junction boxes by the hand pumps shown one on each side just above the collector.

The two distributors or junction boxes feed the engines; that on the left feeding the four bottom engines, whilst the other supplies the two top engines. As the petrol supplied to these distributors is more than that used by the engines, the extra amount passes up to four small gravity tanks which will be seen in the illustration and which are situated in the planes. These hold about 15 gallons each. As the excess of petrol continues the gravity tanks overflow through return pipes into the main collector. In these pipes flow indicators are fitted in order that the engineer may see that the system is properly working. After the main tanks are empty and no more petrol is supplied to the distributors the petrol flows from the gravity tanks into the junction boxes which ensure a further 60 gallons. The two junction boxes are fitted with indicators to show the pressure or head of petrol supplied. They are also fitted with relief valves discharging back into the collector. The collector is also fitted with a large filter, through which all the petrol passes on its way to the engine.

A great deal of progress has been achieved in the general arrangement of the engine and its cylinders so that the most effective use is made of all the material. With cylinders in line the deadweight loss per horsepower will rapidly decrease after the first cylinder, but after about four cylinders the economy to be gained by adding more cylinders practically ceases owing to the increased size of the crankshaft and bearings required to resist torsion. This naturally leads to grouping the cylinders in two rows in V form working on a common crankshaft, with two connecting rods to each crank-pin. Many such engines have been made with two rows of either four or six-cylinders each. Further extension of this principle has enabled Messrs. Napier to produce a twelve-cylinder engine of 450 horsepower, with three rows of four cylinders (three connecting rods to each crank-pin), and weighing only 1.86 pounds per horsepower dry weight.

In addition to placing cylinders in line, designers have from the earliest days of aero engines been attracted by the scheme for mounting all the cylinders radially around a common crank. By this arrangement the length of crankshaft and crank-case is, of course, reduced to a minimum, and there have been engines with three, five, six, seven, nine, ten, and fourteen cylinders in one or two planes. Of this type the Cosmos Company have produced a 450-horsepower nine-cylinder air-cooled radial engine, which weighs only 1.47 pounds per horsepower, or less than one-third the weights of similar type French engines of 1914-15.

Radial engines include those of the rotary air-cooled type, and progress in this type can be best indicated by taking the French 80-horsepower Gnome of 3.26 pounds per brake horsepower, and comparing it with the British Bentley engine of 230 horsepower, weighing only 2.165 pounds per horsepower, a reduction of about 30 per cent in spite of the tremendous increase in the centrifugal load. With radial engines of large size, however, the increased head resistance is a serious objection, and it is not likely that this method of arrangement can be extended indefinitely as regards size.

PARTICULARS OF REPRESENTATIVE BRITISH AERO ENGINES OF EACH TYPE, MAY, 1919.

Type.	Method of Cooling	No. of Cylinder.	Bore m/m	Stroke m/m	Compression Ratio—to 1.	Piston Speed ft. per minute.	B.H.P.		Speed		M. E. P.		Normal Propeller Speed	Dry Weight in Lbs.	Lbs. per H. P.		Petrol Lbs. per Normal H. P. Hour	Oil Lbs. per Normal H. P. Hour
							Normal	Max.	Normal R. P. M.	Max. R. P. M	Normal	Max.			Dry	Wet		
Bentley, Rotary 2 (Fig. 15)	Air	9	140.00	180.00	5 26	1,536.0	230	234	1,300	1,360	92.0	......	1,300	498	2.165	...	0.63	0.088
A. B. C. Fixed Radial, "Dragon-fly" (Fig. 17)..	Air	9	139.7	165.09	4.42	1,787.5	320	350	1,650	1,750	110.0	......	1,650	635	1.98	...	0.585	0.028
Straight Six, Siddeley, "Puma" (Fig. 14)......	Water	6	145.00	190.00	5.00	1,558.0	250	266	1,400	1,500	115.0	123.0	1,400	645	2.58	...	0.6	0.062
2 rows of 4 V. at 90 deg. Sunbeam, "Arab"......	Water	8	120.00	130.00	5.3	1,707.0	212	220	2,000	2,100	112.0	......	1,200	550	2.6	3.24	0.486	0.039
2 rows of 6 V. at 60 deg. Rolls-Royce "Eagle" 8 (Fig. 23)....................	Water	12	114.29	165.09	5.3	1,950.0	359	368	1,800	1,900	127.2	131.4 at 1,500	1,080	926	2.58	3.23	0.5	0.025
2 rows of 6 V. at 60 deg. Galloway, "Atlantic" (Fig. 24)....................	Water	12	145.00	190.00	5.4	1,872.0	550	575	1,500	1,600	126.3	......	1,500	1,150	2.09	2.74	0.504	0.045
2 rows of 6 V. at 60 deg. Rolls-Royce "Condor"	Water	12	139.69	190.5	5.1	2,187.5	610	656	1,750	2,000	129.0	129.4	971	1,350	2.21	2.86	0.495	0.0225
3 rows of 4 V. at 60 deg. Napier "Lion" (Fig. 22)	Water	12	139.69	130.17	5.55	1,708.0	450	468	2,000	2,100	122.0	126.0	840	1,318	1 86	2.51	0.495	0.0225

A saving in weight and great improvement in mechanical efficiency have been obtained by eliminating the valve tappet rods and substituting overhead camshafts operating directly on the valves. Further economies have been effected by the employment of aluminium for the cylinders fitted with either steel or cast-iron liners. Pistons are now universally made of aluminium, and the gain in mechanical efficiency has been of even greater importance than the saving of weight. Hundreds of other details have also been the subject of most exhaustive study and improvement.

The question naturally arises, how is it that such results have been obtained in such a short time and in face of so many difficulties with regard to labor and material. In the first place, the result has been obtained by the free use of money; the question of expense could naturally not be allowed to stand in the way of progress in this all-important matter. Money has been spent on research, on steels and other materials, processes of manufacture, inspection organizations, etc., and the amount of scrapping of material and parts not up to standard has been on a scale that no commercial firm could adopt, and live. The greatest credit is due to the steel manufacturers for the manner in which they collaborated with the Government on the evolution of special steels and the preparation of standardized specifications. The table on page 766 gives the principal particulars of some of the leading engines of today.

PART IV.—NAVIGATION AND METEOROLOGY.

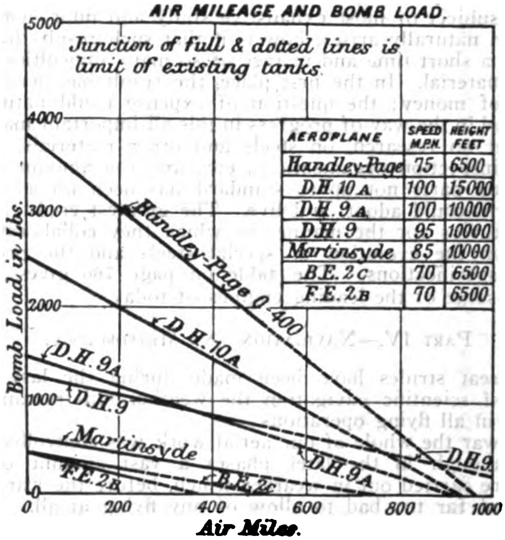
Although great strides have been made during the last year in the development of scientific navigation the weather still remains the dominating factor in all flying operations.

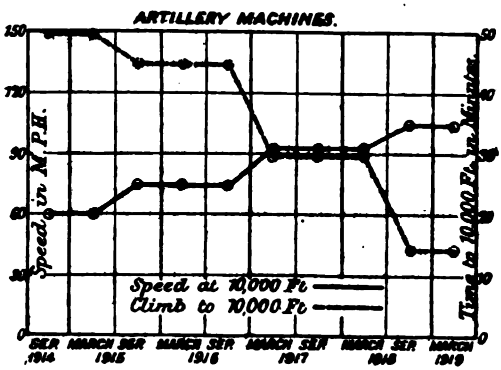
During the war the whole of our aerial work was controlled by weather conditions, although in the later phases a vast amount of successful operations were carried out in weather which, before the war, would have been considered far too bad to allow of any flying at all. The problem of flying in anything like a thick fog or ground mist remains unsolved, and if flying in this country is to be a real commercial proposition much solid research in this direction will be necessary.

When the war started, navigation was dependent mainly on map reading. Our machines were fitted with compasses, but these were of indifferent design, and could not be relied on unless the course could be frequently checked by a direct view of the ground below. The design of an aeroplane compass presents special problems, and the errors involved in using a compass of the type used on ships had not been realized before the war. These problems were successfully investigated by the late Keith Lucas, who evolved a satisfactory aeroplane compass. This compass and subsequent types based on its design were standardized. Even with the improved compass, navigation in unfavorable weather conditions still presents great difficulties.

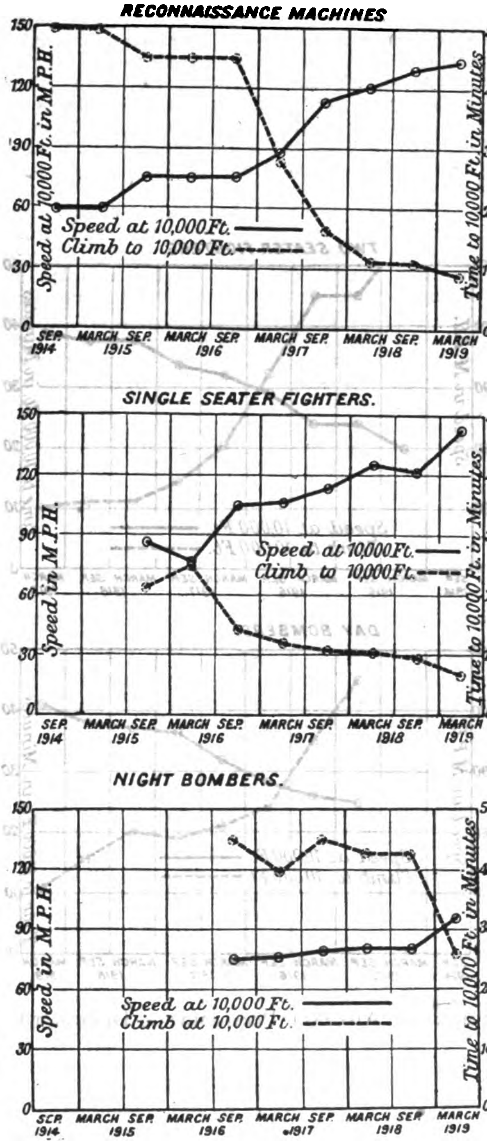
The very great advantages, especially in long-distance bombing operations, of being able to fly for considerable distances through clouds were clearly realized, and a special research was undertaken to make this a practical proposition. Excellent results were obtained, chiefly due to the introduction of an instrument called the turn indicator. When flying in a cloud or a fog bank the pilot loses all sense of direction, but this instrument used in conjunction with the compass enables him to steer a straight course, even in these circumstances.

At the time of the signing of the armistice we were training pilots in large numbers in the use of these instruments, and there is no doubt that the Germans would have had some unpleasant surprises a few months afterwards, when formations of bombing aeroplanes would have suddenly emerged from the clouds over the objectives when the inhabitants below might reasonably have considered themselves safe. Before any attempt

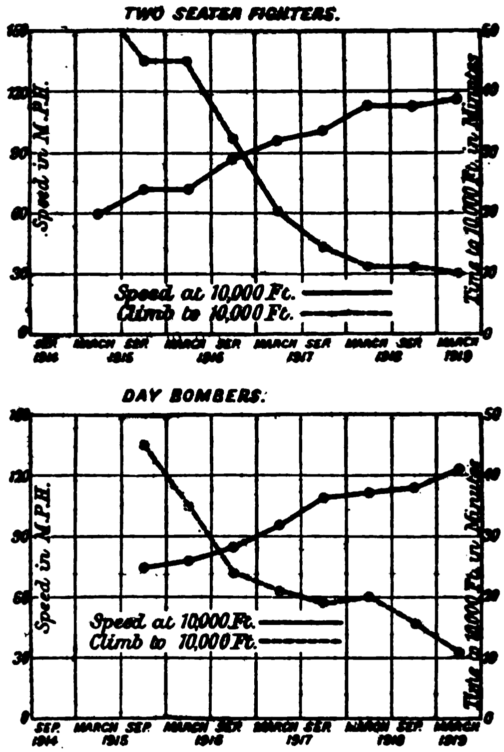




DEVELOPMENT OF AIRCRAFT PERFORMANCE DURING THE WAR.



Development of Aircraft Performance During the War.



Development of Aircraft Performance During the War.

could be made to attack them our aeroplanes would have been back in the clouds again and on their way home.

Navigation by dead reckoning is, of course, dependent on meteorological observations and for us, therefore, a good meteorological service was a *sine qua non.* It must be remembered that on the Western front we had always this important disadvantage, that the prevailing wind was blowing from west to east, thus tending to drift our aeroplanes away from our lines and reduce their chances of getting back when in difficulty. In spite of this the vast majority of air fighting took place well on the German side of the lines, and, even allowing for the German raids over England, the amount of reconnaissance and bombing we carried out over German occupied territory was far greater than that attempted by them on our side of the line. Although, of course, the credit for this is mainly due to the magnificent spirit of our *personnel,* the effectiveness of our meteorological service had some effect in establishing our ascendancy.

A few words may be added about the use of directional wireless for navigation. The Germans used this method for directing their Zeppelins in the raid over England, but the system they used involved the sending out of frequent signals from the airship to be picked up by the German wireless stations. The latter would then send out signals to the airship indicating its exact position. This system had the great disadvantage that we also could pick up the airship's signals and thus locate its position. So we always knew where they were coming from, and arranged our defence accordingly. Our own system was for the aircraft to carry the directional coils, and by picking up signals from home stations, to locate its own position. Our apparatus was only perfected late in the war, but we counted on using it effectively in the projected raids over Berlin, to which I have already referred. With further research and development there are good prospects that directional wireless will be of the greatest value in the long distance navigation of commercial aircraft in the future.

Future Developments.—Any reference to the future of aviation must necessarily be very brief and purely general in character. The rapid and quite abnormal growth in the technique and application of aircraft under the stress and artificial conditions of war time must now give place to the more sober rate of development of peace time.

Our peace-time Royal Air Force, as I visualize it, will become a relatively small organization of remarkable efficiency with the highest ideals and with the keenest *esprit de corps.* Its *materiel* should represent the last word in technical progress such as can only be achieved by considerable expenditure. Quality must be the keynote of its policy not only in *materiel* but also in *personnel.*

An outbreak of war must see us with the very best designs of engine and aircraft, tried and tested, and with a manufacturing nucleus on which war production may be readily expanded.

In regard to civil aviation, one is naturally unhappy at the necessity of adopting the prophetic role, but one factor is clear and definite. Our Government must accept the heritage of war experience, and by action, support and sympathy, encourage and develop its translation into the new channels of peace requirements.

The more immediate problems of international and domestic aerial legislation have now been provisionally solved by the International Aerial Convention and by the Civil Aviation Act, and it is very gratifying that in both these directions Great Britain has taken the lead and shown the way.

In another direction much remains to be carried out quickly. Our great mercantile marine in all its technical perfection would avail but little without our harbors, docks, lighthouses, charts, and navigational instruments. We possess fleets of aircraft of reliability and of great performance possibility, but our navigational facilities are still almost non-existent,

and herein lies one of the main fields of action of our new Department of Civil Aviation.

In all considerations for the future of Civil aviation, the two qualities of outstanding merit appertainable to the new form of transport are speed and independence of action as against land transport requiring roads or rails. These constitute the only assets for any commercial future. Speed in transport is associated with high cost, and speed will always command a high value. Early action should be taken in regard to a few main routes, especially in countries with equable weather conditions, and especially in new countries, backward in rail development.

To such main routes would be Egypt to India, and Egypt to South Africa.

Another early development with immediate possibilities will be the use of large flying boats for inter-island and coastal traffic, for example, in the West Indies and New Zealand. The independence of the flying boat in regard to costly aerodromes, and its higher degree of safety in regard to landing places is a factor of great immediate importance. The Government and the large existing transport companies, shipowners and railways might well collaborate in their early efforts, as the initial financial results will not be healthy enough for entirely new enterprises.

Another aspect must not be neglected—the encouragement of the sporting and popular use of aircraft—and it would appear essential that the great number of war aeroplanes surplus to Air Force needs should be distributed at a low price and as early as possible to the many small enterprises and even to individuals so that the community at large may have their present enthusiasm and interest in aviation maintained and encouraged.

In technical development of design our Air Ministry must spend money by development contracts, as private enterprise will not be able to support the developments for some considerable time. This applies in particular to engine development. Attention should also be devoted to improvements in landing carriages and landing problems, also the reduction in fire risks.

In conclusion, it would appear that the future rate of development will depend almost entirely on the permissible rate of expenditure and on the efficiency of the outlays.

I have to express my regret that lack of time has prevented this paper from becoming more than a series of somewhat disconnected notes, but it is hoped that these may prove to record some of the more outstanding factors in one of the many great wonders of the world war.

I have to express my deep debt of gratitude to Lieutenant-Colonel Ogilvie and his partners, Messrs. Bristow, Pippard and Watts, for their invaluable assistance in the preparation of the paper, also to General Seeley, Under Secretary of State for Air, for permission to use the photographs of the different machines, and, finally, to the different aeronautical contractors for permission to use the photographs of the different machines.—"Engineering."

BOOK REVIEW.

NAVIGATION AND NAUTICAL ASTRONOMY, PROF. J. H. C. COFFIN, Eleventh Edition, revised and enlarged, by ELMER B. COLLINS, 259 pages. D. VAN NOSTRAND AND CO., $3.00 net.

Prof. Coffin's "Navigation and Nautical Astronomy" is an old friend. The new edition appears in the Van Nostrand binding uniformly with other nautical text books by the same publishers.

The treatment of the subject is practically the same as that of previous editions, but illustrations and problems conform to the new form of the Nautical Almanac and considerable space is given to the exposition of the use of lines of position; one chapter on the Sumner methods and another chapter on the Marc St. Hilaire and Aquino methods. The figures and explanations are clear and concise.

The paper is absorbent and does not lend itself to the underscoring and marginal references useful in a book to be used for frequent reference.—M. W. T.

SIMPLE RULES AND PROBLEMS IN NAVIGATION, 3d Edition, revised and corrected by BRADLEY JONES, 305 pages. CHARLES H. CUGLE. E. P. DUTTON CO., New York; price, $4.00 net. In this, the third edition of Mr. Cugle's book, many of the problems which appeared in earlier editions are worked out, while in addition there are examples for practice, with their answers. Several additional features have been embodied in the present edition. It is an excellent work, with theory subordinated to practical utility.

ASSOCIATION NOTES.

The following members and associates have joined the Society since the publication of the last number of the JOURNAL.

MEMBERS.

Byrne, A. C., Lieutenant, U. S. N.

Enders, John J., Lieutenant, U. S. N.

Hepburn, A. J., Captain, U. S. N.

Kengle, L. F., Lieutenant, U. S. N.

ASSOCIATES.

Allan, John B., Messrs. Cammell Laird & Co., Sheffield, England.

Barnes, Henry, 204 Van Noyes Hotel, Tacoma, Wash.

Dillon, Edward H. J., Dist. Mgr., Griscom-Russell Co., 1636 Land Title Building, Philadelphia, Pa.

Dimm, Wayne T., Newport News Shipbuilding Co., Newport News, Va.

Malseed, James S., Thacher Propeller and Foundry Co., Albany, N. Y.

Pabodie, Robt. J., M.E., Dist. Mgr., Griscom-Russell Co., 836 Monadnock Building, Chicago, Ill.

Sadtler, Edwin B., 233 Broadway, New York City.

Stack, Michael, Marble Court Apartment, 637 Bergen St., Brooklyn, N. Y.

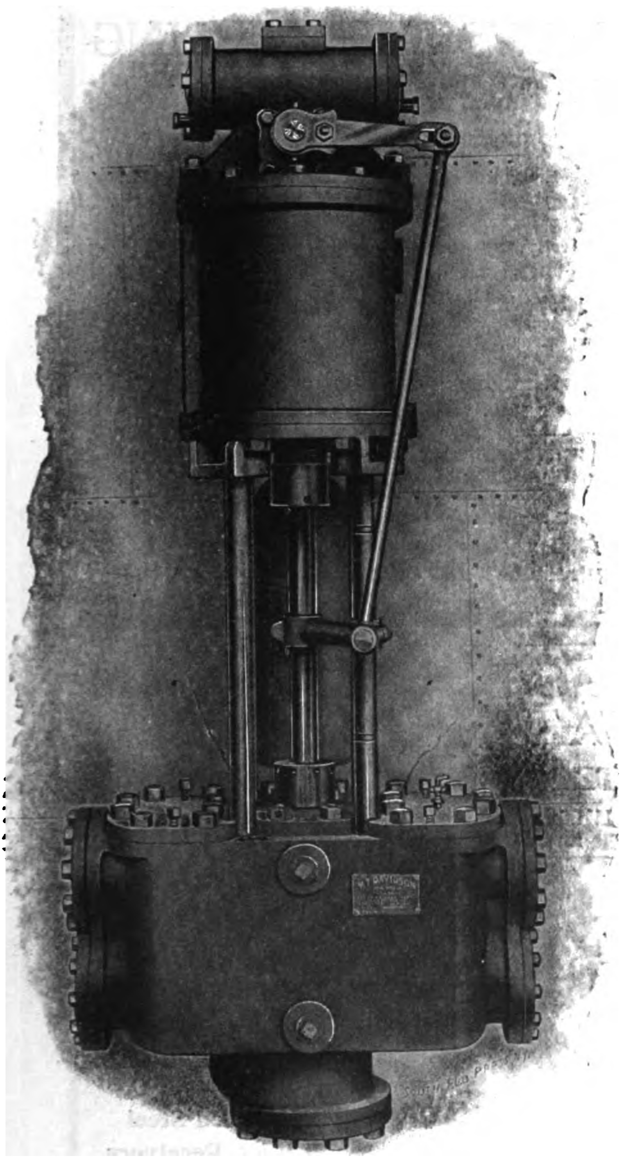
Thacher, Thomas O., Thacher Propeller and Foundry Co., Albany, N. Y.

MEMBERSHIP.

It is desired to increase our membership as much as possible, and the assistance of members and associates is solicited. It is especially requested that heads of Engineer Departments of vessels bring to the attention of younger officers the advantage to be gained by membership.

ADDRESSES.

Do not fail to keep the Secretary advised of your address at all times.



DAVIDSON STEAM PUMPS

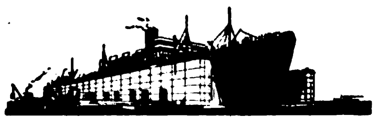
FOR ALL SITUATIONS.

CONDENSERS, EVAPORATING and DISTILLING APPARATUS.

U. S. S. "Connecticut",
"Washington",
"St. Louis",
"Denver",
"Chattanooga",
"Bancroft",
"Baltimore",
"Cleveland",
"Galveston",
"Iris",
"Rainbow",
"Arkansas",
"Bagley",
"Dale",
"Rodgers",
"Stewart",
"Worden",
"Winslow",
"Gloucester",
&c., &c., &c.

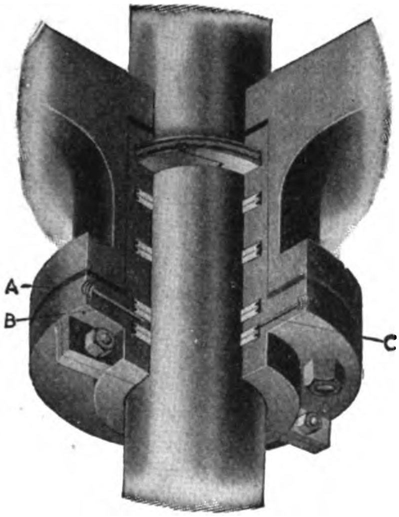
M. T. DAVIDSON COMPANY

43-53 Keap Street,
BROOKLYN, NEW YORK.



FRANCE METALLIC PACKING

FOR ALL CLASSES OF MARINE SERVICE

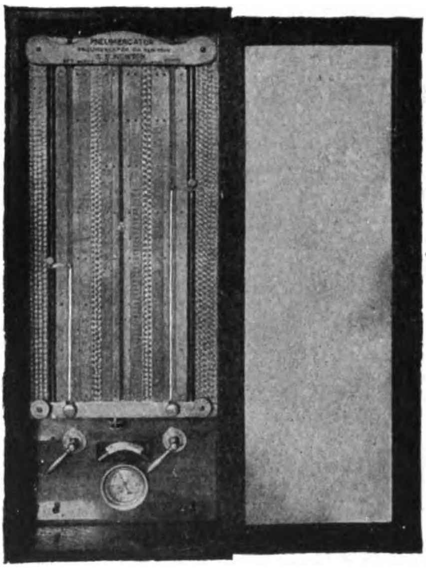


SEND FOR CATALOG

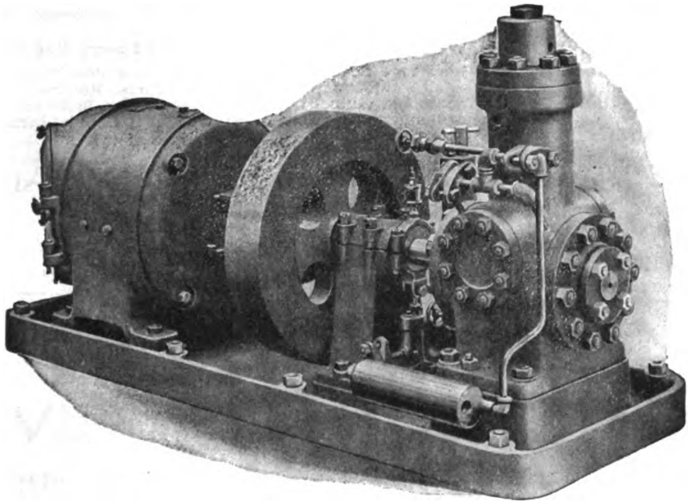
FRANCE PACKING COMPANY

TACONY—PHILADELPHIA, PA.

Model T-1



Model D-1



CO_2 Compressor direct connected to Electric Motor.

REFRIGERATING MACHINERY

Following the {Ammonia Compression / Ammonia Absorption / CO_2 Compression} Principles.

A Machine to meet every Refrigerating Requirement.

In Capacities ranging from 1/8-ton refrigerating duty upwards.

Actuated by any available source of power.

Your Refrigerating Machinery Requirements can best be supplied by the Largest Manufacturers of this class of machinery in the world.

YORK MANUFACTURING CO.

(Ice Making and Refrigerating Machinery exclusively)

YORK, PA.

Consultation on any Refrigerating Problem invited.

TOBIN BRONZE

TRADE-MARK "REGISTERED IN U. S. PATENT OFFICE."

NON-CORROSIVE IN SEA WATER. **Readily Forged at Cherry Red Heat.**

Round, Square and Hexagon Rods for Studs, Bolts, Nuts, etc., Pump Piston Rods, Yacht Shafting, Rolled Sheets and Plates for Pump Linings, Condensers, Rudders, Center Boards, etc. Hull Plates for Yachts and Launches, Powder Press Plates, Boiler and Condenser Tubes. For tensile, torsional and crushing tests see descriptive pamphlet furnished on application.

SOLE MANUFACTURERS

THE AMERICAN BRASS COMPANY

ANSONIA BRASS & COPPER BRANCH

ANSONIA, CONN.

Also Manufacturers of Copper and Brass Sheets, Rods, Tubes and Wire, Etc., Etc.

MASON

REDUCING VALVES

AND OTHER

PRESSURE REGULATORS

answer every requirement for Marine Service where the accurate and automatic control of Steam, Air and Water Pressure is required.

MASON REGULATOR CO.

BOSTON, MASS.

F. J. SCHMIDT CO.

Naval and Army Outfitters

High-Grade Uniforms, Equipments and Civilian Dress

63 MARYLAND AVENUE, ANNAPOLIS, MD.

C. & P. 'PHONE, ANNAPOLIS 241

A Moving Picture of the Heart of a NOKUT Marine Valve

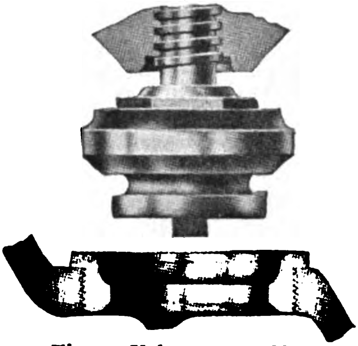


Fig. 1. Valve open. Note large free opening.

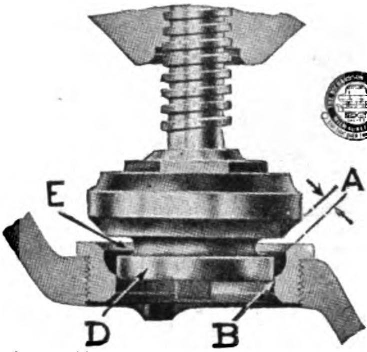


Fig. 2. The valve partly open. The protector throttles the flow

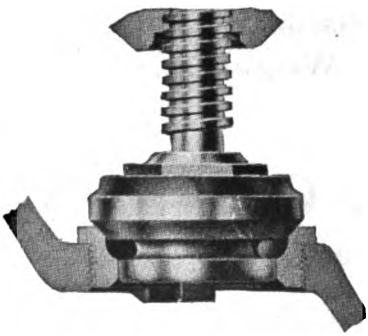


Fig. 3. The valve closed. The unscored seats form a leak-proof joint.

These three consecutive views show the R-P NOKUT valve in the act of closing. Note in Fig. I the large free opening when the valve is wide open.

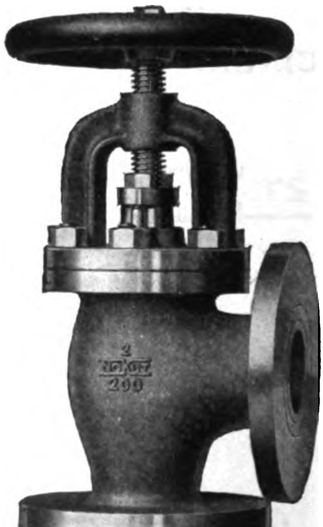
See the details in Fig. 2. "E" is the pressure reducing chamber. "D" is the protector. "A" shows the large opening between seats as compared to the small inlet "B" between the seat ring and the Protector.

The Protector holds back the flow and the Pressure Reducing Chamber reduces the velocity across the seats. Wear or wire drawing thus comes on the Protector, leaving the seats unharmed and tight.

Even did the NOKUT valve cost twice as much as other good valves, it would be an excellent investment, but the NOKUT valve costs no more than other good valves. It is worth your immediate investigation.

THE RICHARDSON-PHENIX CO.
NOKUT Valve Department
WORKS- 114 RESERVOIR AVE. MILWAUKEE, WIS.

Bulletin 100 describes the R-P NOKUT valve. A copy is yours for the asking. Drop us a postal for it today.



PATENT ATTORNEY AND ENGINEER

RAY B. WHITMAN

Bach. of Mar. E., U. of Mich. Mem. Soc. N. A. & Mar. E. Assoc. Mem. A. S. M. E.

505 Fifth Ave., New York

Specialist in marine cases. Patents obtained, including all legal work; practical inventions redesigned, developed and marketed. A peculiarly able, complete and personal service, and on reasonable terms.

Carry More Cargo and Less Engine

De Laval 3 to 1 Reduction Gear for Driving Propeller from Diesel Engine 1,500 H. P.

THE POWER of an internal combustion engine increases almost in direct ratio with the speed. By trebling the speed, the amount of power obtainable from a given weight of engine is also multiplied three-fold.

THE DE LAVAL Double-helical Speed-reducing Gear makes this increase in engine speed possible. It is the result of over 25 years of development in stationary and marine power plants. Its design and methods of production have been perfected by us and De Laval Gears are giving entirely reliable and satisfactory results in marine service.

THE BETTER engine speed and better propeller speed obtained through the use of De Laval Speed-reducing Gears also increase the efficiency of engine and propeller. The efficiency of the gear itself is above 98 per cent.

OUR ENGINEERS would be glad to confer with you on any marine propulsion problem. Ask for Special Publication No. 95.

DE LAVAL STEAM TURBINE CO.

435 NOTTINGHAM WAY TRENTON, N. J.

VOL. XXXI. NO. 4.

JOURNAL

OF THE

AMERICAN SOCIETY

OF

NAVAL ENGINEERS.

PUBLISHED QUARTERLY BY THE SOCIETY, FOR THE ADVANCEMENT OF NAVAL ENGINEERING.

NOVEMBER, 1919.

WASHINGTON, D. C.

R. BERESFORD, PRINTER, 605 F STREET, N. W.

Entered at the Post Office at Washington, D. C., as Second Class Matter.—Acceptance for mailing at special rate of postage provided for in Sec. 1103, Act. of Oct. 3, 1917, authorized on July 3, 1918.

OFFICERS OF THE SOCIETY.

President:

Rear Admiral H. P. NORTON, U. S. Navy.

Secretary-Treasurer:

Commander J. S. EVANS, U. S. Navy.

Council:

Rear Admiral H. P. NORTON, U. S. Navy.
Rear Admiral T. W. KINKAID, U. S. Navy.
Captain ROBERT STOCKER, (C. C.) U. S. Navy.
Commander S. M. ROBINSON, U. S. Navy.
Commander W. W. WHITE, U. S. Navy (Retired).
Captain of Engineers W. M. PRALL, U. S. C. G.
Commander J. S. EVANS, U. S. Navy.

SUBSCRIPTION RATES.

The subscription price of the JOURNAL, postpaid to the United States and possessions, Canada, Cuba and Mexico, is $5.00. Single copies, $1.30. To other countries in the Postal Union, $5.40. Single copies $1.35. All subscriptions are payable in advance.

Make checks, drafts and postal orders payable to

AMERICAN SOCIETY OF NAVAL ENGINEERS.

Advertising rates will be furnished on application.

It is earnestly requested that prompt information be given of changes in address, or of failure to receive the JOURNAL.

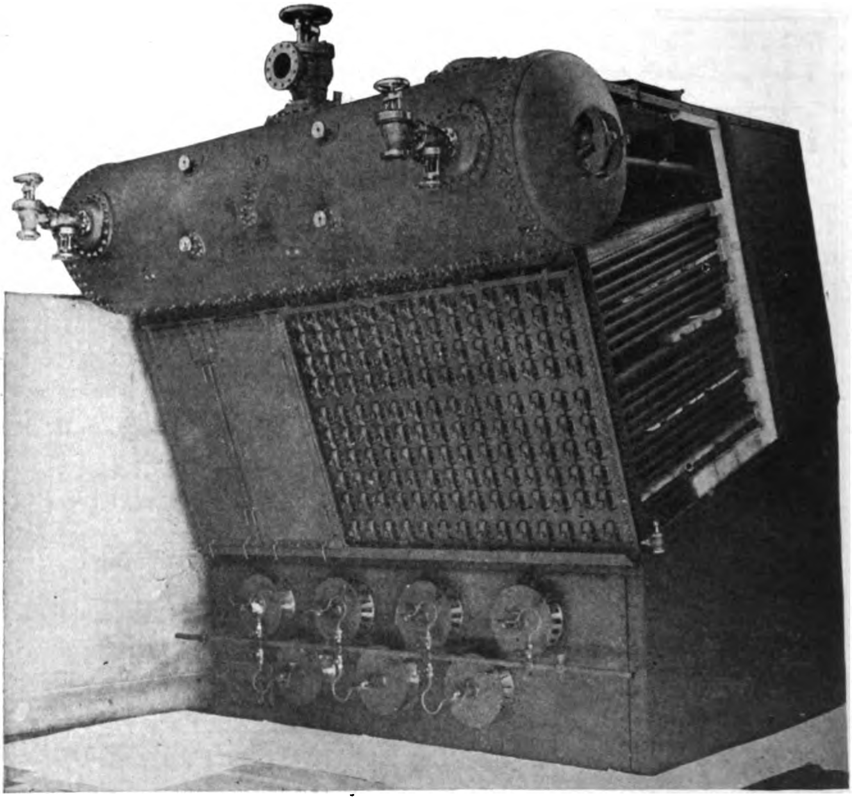
Address all communications for the Society to

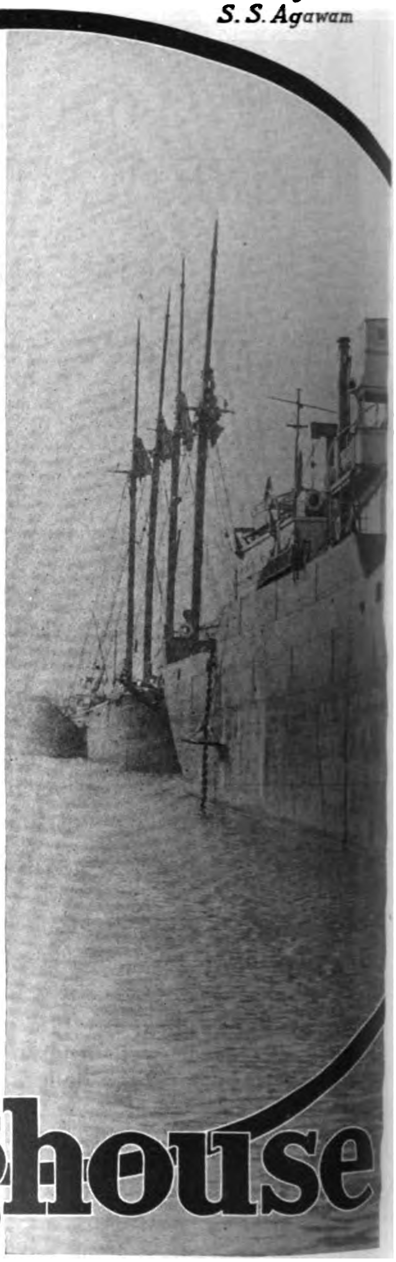
AMERICAN SOCIETY OF NAVAL ENGINEERS,
Navy Department, Washington, D. C.

CONTENTS.

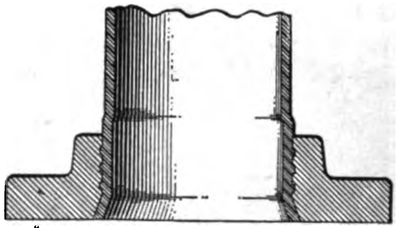
INDEX TO ADVERTISERS.

Name.	Manufacturers of or dealers in	Post office address.	Page
S			
Seattle Construction and Dry Dock Co.	Shipbuilders, Engineers, Boiler Makers, Founders.	Seattle, Wash.	xxxi
Sinclair Refining Company	Quality Petroleum Products	Chicago, Ill.	i
Sterling Engine Company	Sterling Engines	1274 Niagara st., Buffalo, N. Y.	xxxi
Sugar Apparatus Manufacturing Co., Wilmington, Del. Wheeler Condenser & Engineering Co., Carteret, N. J.	Evaporating Apparatus	Carteret, N. J.	xvi
T			
Terry Steam Turbine Co.	Steam Turbines	Hartford, Conn.	xx
Texas Company, The	Texaco Lubricants, Texaco Ursa Oil, Texaco Cylinder and Machine Oils, Texaco Fuel Oil.	17 Battery Place, New York City.	xiv
U			
United States Metallic Packing Co.	Piston Rod and Valve Stem Packings	427 North 13th st., Philadelphia, Pa.	xxx
V			
Vibration Specialty Co.	Vibrations	Harrison Bldg., Philadelphia, Pa.	xi
Volz, William E., M. E.	Positive Patent Lifting Clamp	6 Church St., New York City	xxvii
W			
Ward Engineering Works, The Charles.	Patent Marine Water Tube Boilers	Charleston, W. Va.	i
Warren Steam Pump Co.	Pumps and Condensing Apparatus for Marine Installations.	Warren, Mass.	xxx
Watson & McDaniel Co.	Steam Traps, Separators, Reducing Valves, etc.	124 N. 7th st., Philadelphia, Pa.	xv
Westinghouse Electric and Manufacturing Co.	Electric Machinery of all classes	East Pittsburgh, Pa.	iv v
Whitman, Ray B.	Attorney and Engineer	505 Fifth Ave., New York	xlvi
Wheeler, Manufacturing Co., C. H.	Radojet Air Pumps	18th and Lehigh Avenue, Philadelphia, Pa.	xxxvi
Wheeler Condenser and Engineering Co.	Condenser Tubes	Carteret, N. J.	xli
Wickes Boiler Co.	Wickes Marine Boiler	Saginaw, Mich.	xlvii
Worthington Pump and Machinery Corporation.	Blake Pumps for Naval and Merchant Marine.	115 Broadway, New York City.	vii
Y			
York Manufacturing Co.	Ice Making and Refrigerating Machinery exclusively.	York, Pa.	xl

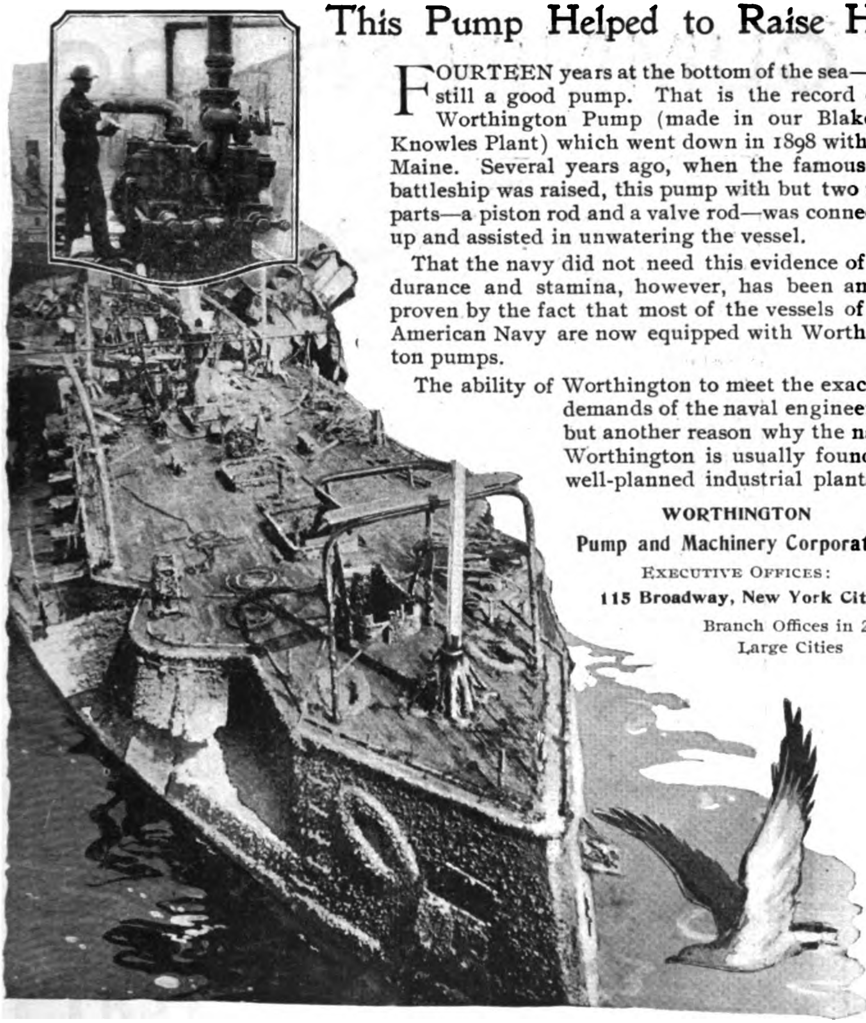




S. S. Agawam



Sunk With the Maine, This Pump Helped to Raise Her



FOURTEEN years at the bottom of the sea—and still a good pump. That is the record of a Worthington Pump (made in our Blake & Knowles Plant) which went down in 1898 with the Maine. Several years ago, when the famous old battleship was raised, this pump with but two new parts—a piston rod and a valve rod—was connected up and assisted in unwatering the vessel.

That the navy did not need this evidence of endurance and stamina, however, has been amply proven by the fact that most of the vessels of the American Navy are now equipped with Worthington pumps.

The ability of Worthington to meet the exacting demands of the naval engineers is but another reason why the name Worthington is usually found in well-planned industrial plants.

WORTHINGTON

Pump and Machinery Corporation

EXECUTIVE OFFICES:

115 Broadway, New York City

Branch Offices in 24 Large Cities

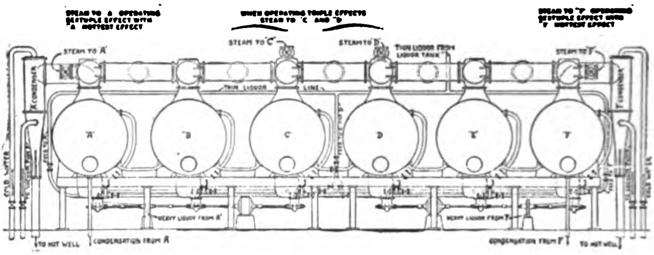
PUMPS-COMPRESSORS-CONDENSERS-OIL & GAS ENGINES-METERS-MINING-ROCK CRUSHING & CEMENT MACHINERY

WORTHINGTON

Deane Works, Holyoke, Mass. Hazleton Works, Hazleton, Pa. Laidlaw Works, Cincinnati, Ohio Gas Engine Works, Cudahy, Wis.

Blake & Knowles Works, East Cambridge, Mass.
Worthington Works, Harrison, N. J.
Power & Mining Works, Cudahy, Wis.
Snow-Holly Works, Buffalo, N. Y.

Two Lillie Triple Effects Converted Into A Sextuple Effect.



IN recent advertisements, we have told about such exclusive Lillie advantages as Film Evaporation, Mechanical Circulation, Reversibility, minimum incrustation troubles, ability to increase the size of each unit, etc.

HERE is another important advantage: The above drawing shows how two old model triple Lillie effects were placed in line and converted into a Vapor Reversing Sextuple Effect.

THIS sextuple effect is operated with steam in the hot effect at about four pounds pressure. It works practically without incrustations on the tubes.

BEFORE the combination was made, the non-vapor-reversing triple effects gave about the heaviest incrustations we have ever seen. Similar incrustations were obtained in a vertical tube double effect when evaporating the same liquors.

THE Lillie is the most efficient of all evaporators. Comparative tests prove that it is far superior in every respect.

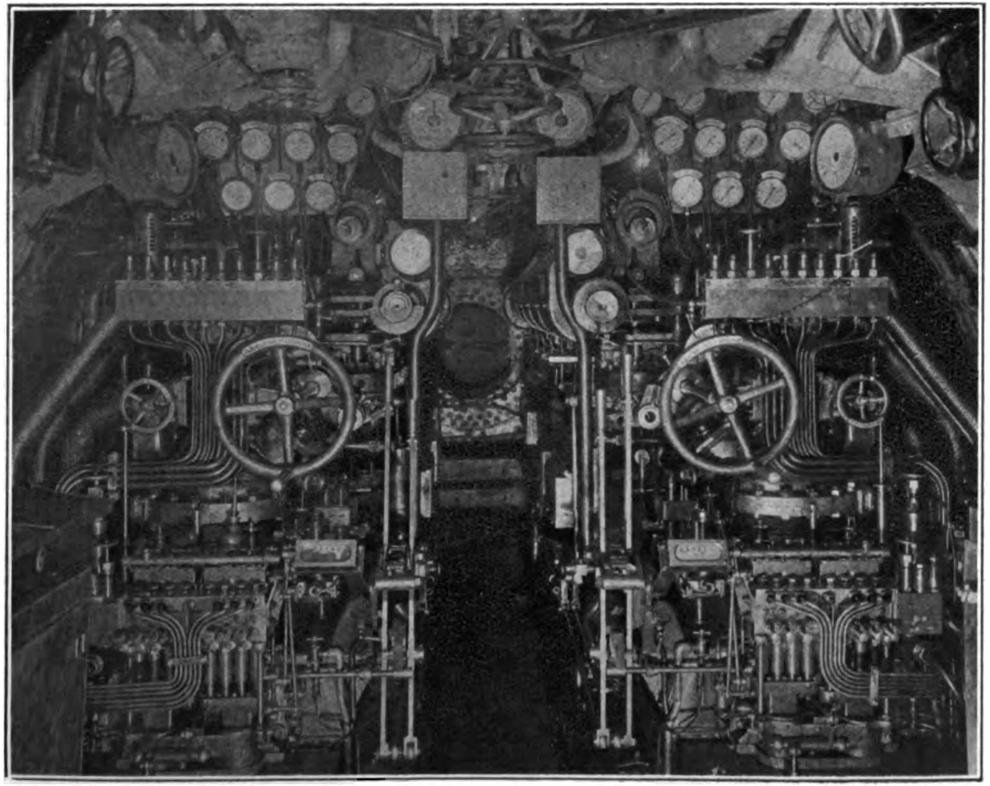
GIVE US YOUR EVAPORATION REQUIREMENTS IN DETAIL.

WHEELER CONDENSER & ENGINEERING CO.

CARTERET, N. J.

Under agreement with the Sugar Apparatus Manufacturing Company—S. Morris Lillie, Pres.—Owners of the Lillie Evaporator patents.

41



ENGINE ROOM OF "U-164."

JOURNAL

OF THE

AMERICAN SOCIETY OF NAVAL ENGINEERS

VOL. XXXI. NOVEMBER, 1919. No. 4.

The Society as a body is not responsible for statements made by individual members.

GERMAN SUBMARINE ENGINES.

BY LIEUT. COMDR. H. T. SMITH, U. S. N., MEMBER.

In view of the prominence attained by the Diesel Engine in Submarine practice, and of its more recent application to an increasing degree in commercial practice, the following brief description of a German Submarine engine obtained from inspection of a number of these vessels is thought will be of interest to the service. The German answer to many questions which have puzzled our designers and in many cases which are still open questions may be found in these engines. Prominent points in this answer are: The four-stroke cycle; air injection; air starting and reversing; high revolutions and piston speed; oil-cooled pistons; extreme rigidity; stroke-bore ratio close to 1.0 or less; and the application of six cylinders to powers up to 1,750 B.H.P. The following table shows the general characteristics of the machinery installation of thirty-six submarines inspected.

Boat.	Make of Engine.	Number in Boat.	Cycle.	Number of Cylinders.	Horsepower Engine.	Revolutions Per Minute.	Air. Starting.	Air. Reversing.	Year.
U-19	M. A. N.	2	4	6	850	450	Yes	Yes	1912
U-53	M. A. N.	2	4	6	1,200	450	Yes	Yes	......
U-55	M. A. N.	2	4	6	1,200	450	Yes	Yes	1916
U-67	Krupp	2	2	6	(1)	(1)	Yes	Yes	......
U-79	Vulcan	2	4	6	500	450	Yes	No	......
U-101	M. A. N.	2	4	6	1,200	450	Yes	Yes	......
U-108	M. A. N.	2	4	6	1,200	450	Yes	Yes	......
U-112	Krupp	2	2	6	1,600	350	Yes	Yes	1917
U-114	Krupp	2	2	6	(2)	(1)	Yes	Yes	1917
U-117	M. A. N.	2	4	6	1,200	(1)	Yes	Yes	......
U-119	M. A. N.	2	4	6	1,200	450	Yes	Yes	1917
U-120	M. A. N.	2	4	6	1,200	450	Yes	Yes	.....
U-122	M. A. N.	2	4	6	1,200	450	Yes	Yes	......
U-124	(3)	2	4	6	1,200	450	Yes	Yes	......
U-125	....	2	4	6	1,200	450	Yes	Yes	1917
U-135	....	2	4	6	1,750	380	Yes	Yes	1917
In addition, this vessel has charging engines as follows:									
....	M. A. N.	2	4	...	450	400	Yes	No	1917
U-139	Krupp	2	2	6	1,600	350	Yes	Yes	1915
Designed for one charging engine, but not installed:									
U-141	M. A N.	2	4	6	1,750	380	Yes	Yes	1917
In addition, this vessel has one charging engine, as follows:									
....	M. A. N.	1	4	6	450	400	Yes	No	1917
U-146	M. A. N.	2	4	6	1,000	450	Yes	Yes	1915
U-153	Krupp	2	4	6	(2)	.....	Yes	No	1916
U-160	M. A. N.	2	4	6	1,200	450	Yes	Yes	......
U-164	M. A. N.	2	4	6	1,200	450	Yes	Yes	1917
UB-99	Vulcan	2	4	6	550	450	Yes	Yes	......
UB-112	M. A. N.	2	4	6	550	450	Yes	Yes	1917
UB-114	M. A. N.	2	4	6	550	450	Yes	Yes	1917
UB-117	Vulcan	2	4	6	550	450	Yes	Yes	......
UB-122	Korting	2	4	6	530	450	Yes	Yes	......
UB-131	Benz	2	4	6	530	450	Yes	......	1918
UB-149	Korting	2	4	6	530	430	Yes	......	1918
UC-94	M. A. N.	2	4	6	300	450	Yes	No	......
UC-98	M. A. N.	2	4	6	300	450	Yes	No	......
UC-101	M. A. N.	2	4	6	300	450	Yes	No	1918
UC-103	M. A. N.	2	4	6	300	450	Yes	No	1918
UC-104	M. A. N.	2	4	6	300	450	Yes	No	1918
UC-105	M. A. N.	2	4	6	300	450	Yes	No	1918
UA-	Krupp	2	4	6	(1)	.....	Yes	No	1913

(1)—Not known (2)—About 1,000 (3)—Blohn-Voss

The above table shows the engine installation of thirty-six boats taken at random; of these five are equipped with two-cycle engines. The two-cycle engines inspected were all of the Germania Krupp type and similar, as far as could be seen, to the Krupp design before the war. The engines of the four-cycle type were manufactured in almost all cases by the M.A.N. Co. The few other makes found so resembled the M.A.N. type as to indicate their manufacture from the M.A.N. design.

The prevalency of the M.A.N. four-cycle engine proved beyond doubt the superiority of this type in Germany. For this reason that type only was examined in detail and furnishes the material for as much detailed information as it was possible to collect for this report. The inspection was concentrated on the engines of *U-164*, which are six-cylinder, four-cycle, air starting and reversing, 1,200 horsepower, at 450 revolutions per minute.

Sizes.—The M.A.N. engine was found mainly in four sizes, namely:

300 horsepower at 450 revolutions per minute,—stroke, 10.937-inch bore.

550 horsepower at 450 revolutions per minute,—stroke, 12.792-inch bore.

1,200 horsepower at 450 revolutions per minute, 17-inch stroke, 17.750-inch bore.

1,750 horsepower at 380 revolutions per minute, 530-mm. stroke, 530-mm. bore.

The exceptions to the above four standard types are the 850-horsepower M.A.N. four-cycle engine of an old model, 1912, and the charging engines on the submarine cruisers *U-135* and *U-141*. These charging engines are the M.A.N. design, 450 horsepower at 400 revolutions per minute, air starting.

All the above types are made in six-cylinder units and are air starting; all but the 300-horsepower type are reversing.

Material.—A shortage of copper is noticeable in the con-

struction of these engines or rather a progressive shortage, for some of the earlier Krupp engines are made of bronze. Steel is used to a very large extent; the cylinders, heads, housings, bedplates and piping, together with other parts which are found in general practice, are of this material. The bedplate and housing should be especially noted in this connection since it is rather a complicated casting with parts as thin as one-half inch or less. Engine parts were not painted, but apparently all unfinished castings were painted with an anticorrosive wash or iron filler paint.

Engine-Room Arrangement.—The general arrangement of the engines and auxiliaries is similar to our practice. The two engines are placed abreast and turn outboard going ahead. The attached lubricating pumps are driven from the vertical shaft at the after end of the engines. The attached circulating pumps are driven off the crankshaft at the forward end of the engines. The two auxiliary lubricating pumps are forward and outboard of the engines, above the floor plates and are motor driven on horizontal shafts. The two auxiliary circulating water pumps are at the forward end of the engine room; the motors are above the floor plates and the pumps below, the shafts being vertical.

The control of the engine is at the forward, inboard side of each engine.

Spares and tools are stowed in metal lockers, built in, outboard of the engines, forward of the engines, under the floor plates, between the engines and in racks overhead and outboard on the hull.

Except in the boats larger than the 800-ton type the spares consist of small engine and auxiliary parts, such as valves, etc. Only on the large boats are cylinder heads or pistons found.

Tools, both standard and special, are neatly stowed in racks with a silhouette of each tool painted on the rack to insure proper stowage. Special tools are all numbered for the purpose intended.

Working room around the engines is limited, particularly abaft and outboard of the engines; however, the engines are generally accessible. Disassembling gear is provided overhead; athwartship tracks and chain falls; but apparently the engines are found to require little or no major overhaul at sea. The floor plates are pressed, bumped steel, are very strong and accurately fitted. They are secured to the angle irons by means of dogs operated by flush head bolts, slotted for a special wrench or large screw driver.

Cylinder Heads.—Cylinder heads are made of cast steel, machine-finished outside. The top and bottom surfaces are practically flat and the horizontal section square with rounded corners. Adjacent heads are held together by means of steel tie plates.

There are six valve openings in the head. This arrangement is varied in the 300-horsepower engines on account of the single-spray valve.

Passages, open to the starting valves, are provided, fore and aft, through the inboard part of the head. These starting air passages are connected between cylinders by jumpers. The whole serves as a starting air manifold.

The arrangement of valve openings and cooling spaces is very symmetrical, allowing of equal expansion under running heat.

Castings are made in rights and lefts in order that adjacent heads may be placed with inlet or exhaust passages adjacent.

The cylinder-head joint is of the usual spigot type, with copper gasket similar to our practice.

Cylinder heads are secured by means of eight steel bolts, 2⅛ inches in diameter.

The indicator cock connection is through the inboard side of the head.

Cylinders.—Engine cylinders are of cast steel with separate liners. The jacket and skirt which form the upper half of

the housing are cast integrally. Webs and flanges are provided on the forward and after sides for bolting to adjacent cylinders and at the bottom for bolting to bedplate flange joints. These are metal-to-metal and the whole construction is very substantial and rigid. On the 300-horsepower engines the working cylinders are cast in pairs. Handholes are provided in the water jackets for cleaning. The cylinder liner is apparently made of close-grained cast iron. It is secured at the top where the head presses it against a shoulder in the cylinder casting; a stuffing box is provided at the bottom of the cylinder jacket to provide for lateral expansion of the liner and at the same time insured watertightness of the water space between jacket and liner. The stuffing box is secured by tap bolts, wired.

Bedplate.—The bedplate is of channel construction, generally similar to our practice, except that the sides are carried up much higher. There is no separate housing provided; the skirts or lower parts of the cylinders and the high sides of the bedplate form this part. Abreast each main bearing, inboard and outboard, there is built up a girder construction connecting the bedplates and cylinders, which gives great vertical strength and athwartship rigidity. This construction, together with the bolted connection between cylinders at the top and the cylinder-head tie-plates, makes the whole engine extremely rigid and free from vibration when running. The bedplates for the 1,200-horsepower engines are made up in three sections with bolted metal-to-metal joints.

Pistons and Wrist Pins.—The working pistons are of the trunk type, and appeared to be made of fine close-grained cast iron. Each piston is provided with five rings at the top, the lower one being a wiper ring, and one wiper ring about 5 inches from the bottom. The piston head is dished, forming a combustion chamber of cylindrical shape with flat top and dished bottom. Piston rings are lap-jointed, and are fitted with dowel pins to prevent rotation. Piston clearance on the

1,200-horsepower engines as taken with feelers was found to be roughly 0.016 inch at the bottom, and 0.062 inch at the top.

The wrist pin is of case-hardened forged steel. It is straight, not tapered. The wrist pin is secured by two large bolts passing through the bosses and piston walls. The securing bolts are in turn secured by set screws.

Valves.—All principal valves are placed vertically in the cylinder heads. The inlet and exhaust are poppet valves of the usual type, the exhaust valve being water-cooled (by a connection to the circulating water system). The inlet valves are of cast steel in one piece. The exhaust valve stems are of cast steel, on which are welded cast-iron heads. A cooling chamber is cored in the exhaust-valve head. Cooling water is forced in through a small tube in the stem and discharged out through the annular space around the supply tube. The lift of the inlet and exhaust valves is about 1.5 inches.

Air-starting valves and relief valves are very similar to our practice. The relief valves are provided with air plungers for positive lifting when reversing the engines.

The spray valves are provided in pairs to each cylinder, except on the 300-horsepower engine. The lift of the valve spindle is controlled to choke the spray air at low speeds. Each spray valve is provided with a nozzle plate drilled with six 2.4-millimeter holes. The internal construction of the spray valve, the atomizer, etc., is according to general practice. One special feature is the actuating connection between rocker arm and the spray-valve spindles. One rocker arm, which is forked on the valve end and carries a tappet plate on trunnions, actuates both valve spindles. Instead of the usual practice of lifting the spindle by direct contact between the tappet and a fixed shoulder on the spindle, in this engine a spring is interposed.

The spray-valve cam begins to take on the roller about 30 degrees before top firing center, but the valve itself does not open until about 8 degrees to 10 degrees before center.

Valve Gear.—The cam shaft is located on the inboard side of the engine at the level of the heads and is driven by worm and gear from a vertical shaft at the after end. The vertical shaft is driven by worm and gear from the crankshaft. The 2:1 gear ratio is obtained in the gears between the vertical shaft and cam shaft. The cam shaft turns inboard going ahead. The cams are solid and keyed on the shaft, except the spray cam, which is an adjustable insert on a carrier keyed to the shaft. Two cams are provided for each valve—one ahead and one reverse. The cam-shaft bearings are carried in supports which bolt rigidly to the flanges where the cylinders are joined together. The same supports also carry the bearings for the rocker shaft, which is located directly over the cylinders. The rocker shaft forms the pivot for all the valve rockers, or rather carries eccentric sleeves on which all the valve rockers pivot. The inlet and exhaust rockers are carried on fixed eccentric sleeves on the rocker shaft so that by turning the shaft the valve rockers are lifted. The spray and air-starting rockers are carried on loose eccentric sleeves on the rocker shaft and are thrown in or out by means of the starting levers. All valve rocker arms are held with their tappets against their respective valve stems, the play always being at the roller end where the sloping surface of the cam will start the lifting motion without striking.

Crank Shaft.—The crank shaft is a hollow steel forging in one piece, including the air-compressor cranks. The main cranks are set at 120 degree angles, Numbers 1 and 6, Numbers 2 and 5, and Numbers 3 and 4 being at the same angle. The air-compressor cranks are at 180 degrees and in phase with cranks Numbers 1 and 6. The sequence of cranks is 1, 4, 2, 6, 3, 5 for the starboard engine and 1, 5, 3, 6, 2, 4 for the port engine (Number 1 cylinder is considered forward). The crank shafts are apparently interchangeable. No thrust bearing appears to be fitted, but the end clearances of all crank webs is very small, probably to provide for that purpose. No balance weights are fitted to the crank webs.

Air Compressors.—Four-stage air compressors are provided on all the types noted except the 300 horsepower, which has a three-stage compressor. The cylinders are made in twin castings and are tandem. Air-compressor cranks are set opposite. The lower pistons are double acting, the first stage compression taking place in both cylinders on the down stroke. The lower pistons are of the trunk type, and second-stage compression is obtained in the annular space around the lower pistons. The third and fourth stage compressions are obtained in the two upper cylinders, respectively, single acting. Air discharge to coolers is screened, but not cooled. Air compressor valves are apparently very reliable and require little overhaul. Engine air compressors are used to charge starting bottles and ship's air.

Fuel Pump.—The fuel pump is practically similar to the M.A.N. design as built by the Nlseco Co., refined in detail. The plungers are vertical and carried on two crossheads driven by opposed cranks on a jack shaft which is driven by worm and gear from the crank shaft. The plungers are free on their stems and can be rotated with the fingers. The suction valves are measuring valves and are controlled by the fuel-control hand wheel and governor. There is also provided cut-out control from the starting levers and by air from the conning tower. The discharge valves are automatic checks, two for each pump. Small hand-priming plungers are provided for each cylinder on the fuel pump. They serve to prime the fuel lines before starting. A fuel reservoir is provided in the pump block, fitted with needle float valve and vent cock.

The fuel pump block is provided with eight pump barrels, the two middle ones being for oil (lubricating) pressure to the spray air regulator.

Governor.—The governor is of the simple inertia type and is driven off the forward end of the crank shaft. The inertia weights are small and the whole apparatus compact. It is

apparently an excess speed cut-off governor rather than a speed-control device.

Clutch.—The engine clutch is of the friction type and is air operated. The tail clutch is similar in design, but is operated by hand through worm and gear.

The outer casing or female member of the clutch is carried on the engine shaft, serving also as a flywheel. It is made in two parts, bolted together, and machined to cone surfaces inside, the cones being base to base.

The male part is carried on the motor shaft. The motor shaft carries a flange or spider on which are fixed four large pins parallel to the shaft axis and extending through the spider on either side. On these pins are carried two circular blocks, one forward and one abaft the spider. The blocks have a small axial freedom on the pins. The peripheral surfaces of the blocks are coned to conform to the inner surfaces of the fly wheel. Forcing the blocks apart engages the male and female conical surfaces. Drawing them together releases the surfaces and disengages the clutch. The cone blocks are operated from a sliding sleeve on the motor shaft by a series of links and toggles. The forward link of the system is a heavy steel spring. The conical surfaces are cut with oil grooves. The sleeve on the motor shaft is operated by means of a strong-back, fixed at one end and carried by the piston of an air cylinder at the other. Connection to the clutch sleeve is made by yoke and collar about midway of the strong-back.

Auxiliary operating gear is provided in a small hand air pump and air bottle.

The working parts of the clutch are all machine finished and finely fitted and its action is extremely smooth.

This apparatus was of special interest due to the compactness and small size for the power transmitted—1,200 horsepower.

Exhaust and Muffler.—The exhaust header is made up in three sections of steel with welded seams and flanges. It is

water-jacketed, with the usual jumpers around flanges. Exhaust from the cylinders is taken off by three elbows, each one serving two cylinders. The exhaust header is connected at its after end by a large water-jacketed pipe to the evaporator. From the after side of the evaporator another water-jacketed connection leads through the inboard exhaust valve to the elbow deck connection. At the after end of this elbow outboard is placed a large gate valve, and thence a long section of water-jacketed piping to the muffler. The muffler is a large cylindrical nonpressure expansion chamber. It is baffled by about six transverse plates with semicircular segments cut in their periphery. Segments in adjacent plates are staggered. The muffler is completely water-jacketed, but no water spray is put into the exhaust. The muffler is provided with a number of flapper vents, which are operated from inside the boat. A muffler drain is also provided with valve and reach rod into the motor room. It was noted that the entire exhaust system from the forward end of the exhaust header to the after end of the muffler is water-cooled. The inboard exhaust valves are large water-jacketed globe valves, with loose disks and auxiliary stems for grinding in without disassembly. These valves close against the sea pressure. The outboard exhaust valves are water-jacketed gate valves, situated outside the hull and operated from inboard by a system of reach rods. These are apparently meant to be quick closing.

Thrust Bearings.—Ball-bearing thrust bearings are installed on the line shafts. The installation is small and compact. The casing is made oil-tight, providing an oil bath for the bearing to run in.

Cam Shaft.—The cam shaft is a steel forging. It is carried in bearings whose pedestals are bolted to the intercylinder webs and flanges. The bearing structure is very rigid. The cam shaft rotates in the opposite direction from the crankshaft. It has an axial motion of about 1½ inches. Two cams for each valve are keyed to the cam shaft—one for ahead and

one for reverse. The axial motion of the cam shaft is obtained at the forward end by means of yoke and collar connection to the reversing gear. This motion is accommodated at the drive or after end by the shaft sliding in the drive gear wheel. Two diametrically opposite feathers of large size on the cam shaft, which fit in corresponding slots in the drive gear wheel, transmit the rotary motion.

Rocker Shaft.—The rocker shaft is a hollow steel forging which forms the pivots for all valve rocker arms. It is mounted in bearings carried by the same pedestals as the cam-shaft bearings. This shaft is made in sections for accessibility in lifting cylinder heads—six sections joined at the bearing supports.

The rocker shaft has a rotary motion of about 90 degrees for lifting the inlet and exhaust rockers off their respective cams when sliding the cam shaft on reversal. These valve rockers are pivoted on fixed eccentrics on the rocker shaft.

Carried on free eccentrics on the rocker shaft are the valve rockers for the spray and air-starting valves. These eccentrics are operated by the two starting levers, one controlling the three forward cylinders, the other the three after cylinders. Rotation of these eccentrics throws in the spray or starting rockers as desired. The middle position of these eccentrics leaves both valve rockers off their cams to permit translation of the cam shaft.

Air Starting Control.—For purposes of starting on air the engine is divided into two groups of three cylinders each with a control lever for each group. Both groups or the after group alone may be used for starting. The forward group only may be cut in one fuel while one group is on air. Actually operating the engines in the following practice was followed:

First step—both groups on air starting.

Second step—forward group on fuel.

Third step—after group on fuel.

These steps followed one another as fast as the starting levers could be operated, starting being accomplished in about four or five seconds.

The control lever for the after group: In the starting position—opens the relay of starting air valve, and also puts the eccentrics on the rocker shaft in the position where the starting-valve rockers are operative. It also operates the spray-spindle control to choke the spray air to the first group of cylinders. In the stop or mid-position—starting air valve is closed and the rocker shaft eccentrics are in mid-position with both spray and air-starting valve rockers clear of the cam shaft. In the running position—starting air valve is closed and the rocker-shaft eccentrics are in position to make the spray-valve rockers operative.

The control lever for the forward group: In the starting position, by means of a cam and small rocker shaft it holds down the fuel-measuring valves and cuts fuel off all cylinders. It also puts the rocker-shaft eccentrics in position where air-starting valve rockers are operative; in the stop or middle position it holds the rocker-shaft eccentrics in mid-position, so that both spray and starting-valve rockers are clear of the cam shaft; in the running position it sets the rocker-shaft eccentrics in position making the spray valve rockers operative, and releases the fuel-measuring valves.

The starting levers are interlocked with the reverse gear so that they can not be operated when the reverse gear is in any intermediate position. Conversely, the reversing gear can not be operated when the starting levers are in any but the stop position.

Reversing Gear.—The reversing gear is operated by means of a handwheel on a fore and aft horizontal shaft at the forward end of the engine. By means of a worm and gear, vertical shaft, bevel gears, and link motion, the rocker shaft is rotated to lift the inlet and exhaust rockers clear of their cams. Next by means of a yoke and collar connection the

cam shaft is moved axially about an inch and a half. The third and last step is to roll the rocker shaft back in position so that the inlet and exhaust valves are operative. The above steps are obtained by turning the handwheel eight complete turns. A sliding collar on the handwheel shaft is moved forward by means of a cam and spindle through the shaft while the gear is in any but final ahead or astern positions, and its function is by link connections to brake boxes to lift the cylinder relief valves during reversal.

Spray Spindle Control.—The lift of the spray-valve spindles is limited by a stop which is raised or lowered by means of a connection to the fuel-control handwheel. It operates to choke the spray air at low speeds and powers. Connection is also provided with the starting levers as explained above.

Fuel Control.—Control of the fuel-measuring valves is by means of interconnected links from the fuel handwheel and from the governor substantially the same as our practice. The control operates to hold open the fuel suction or measuring valves during any desired part of the pumping stroke of the plunger. A separate tappet plate is provided to hold the measure-valves completely open by link connection to the starting levers when they are in the "starting" or "stop" position. This tappet plate may also be operated by means of a small air plunger which is controlled by air connection from the conning tower.

Auxiliaries.—The attached oil pump is a multiple impeller pump which operates very quietly. The arrangement of holes in the idler gear and channels in its spindle appear to aid materially in the operation of the pump. It is believed they are arranged to relieve excess pressures where the gears mesh. The pump valves are arranged to accommodate running in either direction. This pump operates at the same revolutions per minute as the engines, being driven off the vertical shaft. The auxiliary oil pumps, two in number, are of similar type driven by separate motors.

The attached circulating water pumps are two-cylinder, single-acting, plunger pumps, driven by opposed cranks on a jack shaft driven by worm and gear from the crank shaft. On the four types of boats inspected these pumps were disconnected. The auxiliary-circulating water pumps, two in number, are of centrifugal type driven by separate motors on vertical shafts.

Fuel Compensation.—No fuel-supply pumps are installed; the fuel is displaced in the tanks by circulating water and forced to the gravity tanks. Compensation is automatic.

Coolers.—Both air and oil coolers are of the straight-tube type, installed with tubes vertical. There are duplicate oil coolers, and coolers for each stage of the engine air compressor. The air coolers on each engine are placed in a single shell, air entering at the top to the respective nests of tubes and passing out through separators at the bottom. Cooling water enters at the sides of coolers and passes across the tubes. It was not ascertained whether or not the water spaces were baffled.

Evaporator and Distiller.—An evaporator or heater is installed in the exhaust line abaft each engine. It is of the straight-tube type, tubes being placed horizontal and athwartships. Exhaust gases pass across the tubes. Evaporator feed is taken from the circulating water system. When not making water, the circulating water is passed through the evaporator coils with vapor pipe secured. The evaporator may be used as a heater for ship's heating and to provide hot water for shower bath. One straight-tube distiller is provided for the two evaporators. It has a separate motor-driven pump for cooling water.

Strainers and Filters.—Strainers are of the basket type, similar to our practice. Filters are provided with muslin bags.

Instruments.—A striking feature of the M.A.N. engine is the permanent equipment of instruments provided. The engines are designed scientifically and installed to be operated

scientifically. Besides the usual equipment of gauges, thermometers are installed to give the temperature of practically every working and vital part of the engine.

A direct reading electric salinometer is installed to give instantaneous measurement of the chlorine content of evaporated water.

An electric apparatus for detecting salt water in the fuel supply is installed.

A differential pressure gauge is installed to indicate the oil pressure to and from the filter. This shows when the filter is restricted and needs cleaning.

Lubricating System.—Sump tanks are installed, located one under the center of each engine. Suction from the sump is taken to either or both attached or separately-driven oil pumps. From the pump the oil is delivered through a twin filter to the cooler. From the cooler the oil is divided between the lubricating and piston cooling systems. The lubricating oil is led to its pressure regulator and from the pressure regulator to a supply manifold pipe located on the inboard side of the crank case. Branches from the supply pipe are taken to each main bearing and to each cylinder. Oil from the main bearings is carried on to the crank-pin bearings and wrist pins from which overflow or leakage drips into the crank case. The lead to the main cylinder delivers oil which is picked up by the lower working piston ring which is fitted as a wiper.

The gear-set at the forward end of the engine is lubricated by connections to the discharge pipe from the pressure regulator.

The gear-set at the after end of the engine is lubricated by connections from the after end of the supply manifold pipe.

The rocker shaft is hollow and has a connection to the supply pipe which lubricates its bearings, the rocker bearings, roller bearings, rolls, and cam-shaft bearings. The cam shaft runs in an oil trough which is provided with drains to the return piston-cooling pipe.

The air-compressor cylinders are lubricated by means of a four-plunger lubricator driven from the countershaft.

Two drains are provided from the crank case to the sump tank, one located at each end of the crank case.

The lubricating oil found is very heavy.

Piston-Cooling System.—As noted above, the piston-cooling oil is branched off the oil discharge from the cooler, the bulk of the oil being taken by this branch. The lead is taken to a supply manifold pipe located along the inboard side of the crank case. Branches are taken off at each cylinder through the housing to the pistons and from the pistons return pipes lead to a sight drain box on the inboard side of the engine over the sump tank. The supply and return connections to the pistons are made up with knuckle joints, very similar to the installation on the *U. S. S. Fulton.*

Thermometers are fitted on the return oil leads to the sight drain box.

A by-pass between the oil-pump discharge and the crank-case drain passes through the piston-cooling oil regulator and serves to regulate the pressure in the cooling system by regulating the effective output of the pump.

Circulating Water System.—The circulating water system conforms to general practice.

Spray Air System.—The spray air system, air starting, and ship's air are interconnected and so used in German practice. The four-stage air compressors deliver air to the tanks at any required pressures up to about 2,500 pounds per square inch. The spray flask also takes the compressor discharge pressure. From the spray flask a lead is taken to the spray air-pressure reducer and regulator and thence to the spray valves. Reliefs are fitted on the spray air lines. The spray air pressure is controlled statically by a spring-loaded reducer, in conjunction with which is an oil-pressure plunger which serves automatically to raise the spray air pressure as the engine speeds up. Pressure for the regulator plunger is obtained from a

two-plunger pump in the fuel-pump block. Lubricating oil is used for a pressure medium.

Fuel System.—Fuel is supplied to the gravity tanks under the pressure of the circulating water system, circulating water being used to displace the fuel oil and thereby also compensate for the weight of fuel used.

The fuel found is very light, about the consistency of kerosene, and smells like kerosene.

Indicator Gear.—Indicator gear is provided for taking power cards. The indicator cock connection is attached to the inboard side of each cylinder head. The stroke motion is obtained from the piston-cooling return link motion by means of a small spindle which passes up through an oil-tight sleeve in the housing.

Air-compressor cylinders are also provided for taking cards.

Workmanship.—The workmanship on the engines is uniformly excellent. The material itself is uniformly fine in structure and shows utmost skill in manufacture. All working parts are machine finished and accurately fitted. The engines are apparently designed in every detail to produce a finished product, each detail contributing to the final result. No make-shift jobs or evidences of afterthought were visible, as is often the case in less scientific machines.

Practically all nuts on the engine are wrench-tight on their bolts or studs the full length of the thread, and all are secured by means of pins, wires, or locking washers. Handwheels on high-pressure air lines are large and fitted with a quarter to a half turn of lost motion on their stems to give a slight striking force for opening or closing. Pipe flanges which require frequent disassembly are made up with portable fastenings. The flanges are slotted in place of drilled bolt holes and the bolts are hinged under the lower flange.

Air bottles are of steel, two spray flasks and two starting bottles being supplied for two 1,200-horsepower engines. The combined capacity of the starting bottles is about 35 cubic feet.

Due to rigid construction and fine balance, the engines operate with extremely little vibration and are comparatively quiet. The engine-room air is free of smoke and gases and the exhaust is perfectly clear and well muffled.

CURRENT-CARRYING CAPACITY OF ELECTRIC CABLES, U. S. NAVY.

CALCULATIONS OF WIRE AND FUSE SIZES.

BY LIEUT.-COMDR. ALEXANDER M. CHARLTON, U. S. NAVY, MEMBER.

When electricity was first introduced in the service the current-carrying capacity of electric cable was specified as not over 1,000 ampères per square inch cross section of conductor. This was equivalent to 1,273 circular mils per ampère of current carried.

Later the specifications were changed to call for 1,000 circular mils per ampère for continuous loading and 500 circular mils per ampère for intermittent loading.

Both of the above specifications were arbitrary assumptions considered to give a factor of safety which would eliminate the possibility of overheating the cables. No consideration was given to initial temperature or to temperature rises. As these specifications were coupled with a percentage of voltage drop which in most cases was the limiting factor, no serious difficulty arose, as the minimum drop allowed usually required a larger cable than was necessary to carry the current. These arbitrary specifications based on experience with comparatively small cables had no scientific basis and as the size of cables increased aboard ship occasional cases of overheated cables were reported.

As will be shown later, we were underloading our cables below the 200,000 circular mil size and overloading them above this size for temperature conditions now considered as safe.

In 1914 the Bureau of Steam Engineering requested the Navy Yard, New York, to conduct tests to determine the current-carrying capacity of Navy standard cables.

In test Number 175, dated December 16, 1915, the Navy Yard reported tests on a number of leaded and armored cables taken from stock, the following sizes being used:

Single Conductor.	*Twin Conductor.*
9,030 circular mils.	4,494 circular mils.
75,850 circular mils.	9,030 circular mils.
98,820 circular mils.	14,350 circular mils.
198,860 circular mils.	22,820 circular mils.
521,970 circular mils.	49,020 circular mils.
	59,940 circular mils.

The cables tested were suspended in a horizontal position four inches from the floor, exposed on all sides to the air. The cables, which were from twenty to thirty feet long, with copper terminals soldered to the ends, were tested one at a time, the ends being connected directly to a motor generator set.

The tests were made for a temperature rise of 30 degrees C. commencing with a room temperature of 25 degrees C. (77 degrees F. to 131 degrees F.). The following results were obtained:

Size of cables, c. m. s.	Ampères rating (Single Conductor)	Ampères rating (Twin Conductor)
4,494	.....	26
9,030	44.5	40
14,350	.....	57
22,820	.....	74
49,020	.....	111
59,940	.....	128
75,850	178.0	...
98,820	225.0	...
198,860	325.0	...
521,970	656.0	...

Plotting these results we get a curve which varies considerably from the Navy specification of 1,000 circular mils per ampère. As will be seen later, the initial temperature used in this test was considered by the Bureau too low, and the allowed temperature rise too great for conditions aboard ship. Also, no grouping and location factor was allowed for. The values obtained by New York, however, agree closely with those in the standard table when reduced to the same limits.

The current which a cable will carry without reaching a temperature dangerous to the copper or insulation depends upon the establishment of a balance between the heat generated by the flow of current through the conductor and the heat conveyed from the conductor to the insulation and thence radiated to the atmosphere. The area over which the dissipation of heat must take place depends upon the circumference of the cable, that is, it is a function of the diameter. The heat generated varies as the square of the current, and inversely as the square of the diameter of the cable.

Resistance $= \rho \frac{1}{a}$ and $a = \frac{\Pi d^2}{4}$. We would expect then that the current allowed for a certain temperature rise would vary as the diameter to the 3/2 power. In consequence the ampèrage per circular mil of cross-section must be less for the large size conductors than for the small ones in order to maintain the same temperature rise.

The formula worked out by the New York Navy Yard for this particular test is

$$I = .015 \sqrt{\frac{d^3}{x}} \text{ where}$$

$d =$ diameter of copper in mils

$x =$ thickness of insulation including tape and braid

$.015 =$ constant depending on temperature conditions, conductivity of insulation, etc.

Further tests by the Navy Yard, New York, gave additional formulæ arranged in this form:

$$I = .01483 \sqrt{\frac{d^3}{x}} \quad 30° \text{ C. rise. Final temperature, } 63° \text{ C.}$$

$$I = .0121 \sqrt{\frac{d^3}{x}} \quad 20° \text{ C. rise. Final temperature, } 65° \text{ C.}$$

$$I = .01373 \sqrt{\frac{d^3}{x}} \quad 26\ 2/3° \text{ C. rise. Final temperature, } 70° \text{ C.}$$

These formulæ were derived as follows:

Watts loss per foot length of cable $= \frac{I^2 r}{d^2}$

r = resistance in ohms per mil foot at final temperature.

d = diameter of copper in mils (for standard cable use diameter of solid wire of equivalent area).

I = current in ampères.

Watts radiated per foot $= \frac{12\ \Pi d}{1000} \times \frac{Tk}{x}$

T = temperature rise in degrees C.

x = thickness of insulation including tape and braid.

k = constant depending on the thermal conductivity of the insulation and the radiating capacity of the outer surface of the insulation or armor.

The average value of k from test Number 175 is .00243.

Equating these two

$$\frac{I^2 r}{d^2} = \frac{12\ \Pi d}{1000} \frac{Tk}{x}$$

$$I = \sqrt{\frac{12\ \Pi T k}{1000\ r} \frac{d^3}{x}}$$

It was realized that the above formulæ were not in strict accordance with theory, but they were more convenient than the purely theoretical formulæ and were considered to give

results which were conservative and within the limits to which the radiation constants could be determined.

At the request of the Navy Yard, New York, Professor Slichter, of Columbia University, checked up the figures of the Navy Yard by comparison with data prepared by Powell and by Langmuir and Dushman.

Professor Slichter stated that the figures proposed by the Navy Yard "— are very reasonable and in accordance with results obtained by other investigators."

At the outbreak of war, the Standards Committee of the American Institute of Electrical Engineers offered its services to the Government. The Bureau of Steam Engineering requested consideration of A.C. high-tension cables proposed for electric propulsion, and subsequently suggested to the committee that the problems of current-carrying capacity of low-tension cables, limiting temperatures and insulation characteristics, could be advantageously considered by the Standards Committee and results obtained which could be used as a standard for the Naval service.

The sub-committee on wires and cables of the Standards Committee undertook the consideration of these problems. The sub-committee was composed of representatives of the Bureau of Steam Engineering and the Navy Yard, New York, and representatives of the following companies, among others:

The Standard Underground Cable Co.
The Habirshaw Electric Cable Co.
The Safety Insulated Wire and Cable Co.
The Simplex Wire and Cable Co.
The General Electric Co.
The Westinghouse Electric and Manufacturing Co.
The New York Edison Co.
The Public Service Corporation of New Jersey.

The report of the committee on D.C. cables was prepared by Mr. Philip Torchio, Chief Electrical Engineer of the New York Edison Co.

The Bureau requested ratings based on the current-carrying capacities of cables under three conditions:

I. Continuous current rating.
II. Thirty-minute rating—starting from the end of the continuous rating run.
III. Thirty-minute rating—starting from room temperature.

In order to simulate conditions aboard ship, all ratings were requested to be based on an ambient temperature of 45 degrees C. (113 degrees F.). This meets fire and engine room requirements in a measure and also takes care of tropical temperatures.

The allowable temperature rise for the various cables under different ratings is as follows:

	Temperature Rise.	
	Rubber Insulation.	**Varnished Cambric Insulation.**
I. Continuous rating......	15° C.	30° C.
II. Thirty-minute rating after continuous run..	25° C.	30° C.
III. Thirty-minute rating starting from room temperature	25° C.	30° C.

Due to the high initial temperature selected as a basis, the temperature rises were necessarily kept low in order to avoid reaching temperatures where the insulation would begin to be affected. The final temperature for varnished cambric insulated and for reinforced rubber insulated cable is 75 degrees C. (167 degrees F.). As these types of cable are specified for use in machinery and boiler spaces, the probable maximum temperatures are provided for.

The values for the various ratings were obtained from the following formulæ:

I. CONTINUOUS RATING.

The energy lost in the cable $= I^2r$ watts. This multiplied by $(h + J)$ equals the temperature rise or $T = I^2r\ (h + J)$

$$\text{and } I = \sqrt{\frac{T}{r\ (h + J)}}.$$

To allow for the lack of heat dissipation due to the proximity of other conductors, location of conductors close to decks, etc., it was decided to limit the current to 80 per cent of the maximum current-carrying capacity, and

$$I_c = .8 \sqrt{\frac{T}{r\ (h + J)}}.$$

Where I = maximum current

I_c = maximum allowable current

T = permissible temperature rise (15 degrees C. for rubber, 30 degrees C. for varnished cambric).

r = resistance in ohms per inch of conductor

h = heat resistance from copper to armor in degrees C. rise per watt per inch of conductor

J = heat resistance from armor to room temperature in degrees C. rise per watt per inch of conductor

.8 = grouping factor considering that four cables may be racked together in a location with poor ventilation such as under deck or adjacent to bulkheads.

The factor 'h' above is expressed by the general equation

$$h = \frac{K}{2\Pi} \log_e \frac{b}{a} + \frac{K^1}{2\Pi} \log_e \frac{c}{b} \text{ etc.}$$

Where a, b, c, d are the diameters of the various belts of insulation and K, K^1, etc., are constants representing the heat resistivity of the various materials in degrees C. rise per watt per inch cube of the materials.

This formula may be derived as follows:

$$\text{Resistance} = \text{specific resistance} \times \frac{\text{length}}{\text{area}}.$$

For a unit of cable the length of path of heat flow from the inside out $= dx$. The area of this path $= 2\Pi x$

$$h = \int_{\frac{a}{2}}^{\frac{b}{2}} \frac{K dx}{2\Pi x}$$

When $b = a$, resistance $= o$
Therefore $C = o$ and

$$h = \frac{K}{2\Pi} \log_e \frac{b}{a} \text{ etc.}$$

II. THIRTY-MINUTE RATING—STARTING FROM THE END OF THE CONTINUOUS RUN.

The energy lost in the conductor in watts in any given time is equal to the watts radiated plus the watts absorbed in the cable due to its thermal capacity.

$$\text{Therefore } \left(\frac{I}{.8}\right)^2 r = \frac{T}{(h+J)} + \frac{w dT}{dt}$$

Where $w =$ the heat input in watt seconds required to raise copper 1 degree C. per second and the insulating materials, lead and armor in proportion to their temperature rise for 1 degree C. increase in copper temperature

$t =$ time in seconds.

$$\frac{dT}{\left(\frac{I}{.8}\right)^2 r - \frac{T}{(h+J)}} = \frac{dt}{w}$$

At the beginning of the thirty-minute run

$$t = o \text{ and } T = \left(\frac{I_c}{.8}\right)^2 r\,(h + J) \text{ and } I = I_c$$

$$\text{Let } P_o = \left(\frac{I_c}{.8}\right)^2 r. \quad \text{Then } T = P_o\,(h + J).$$

At the end of the thirty-minute run $t = 1{,}800$ and $T = T_r =$ maximum allowable temperature.

Integrating, with these limits of integration

$$T_r = (h + J)\left(\frac{I}{.8}\right)^2 r\left[1 - \cosh\frac{1800}{w(h+J)} + \sinh\frac{1800}{w(h+J)}\right]$$
$$+ P_o\,(h + J)\left[\cosh\frac{1800}{w(h+J)} - \sinh\frac{1800}{w(h+J)}\right]$$

III. THIRTY-MINUTE RATING STARTING FROM ROOM TEMPERATURE.

The same differential as in II expresses the relation existing, but in this case the limits are $T = T_r$ and $T = o$ and $t = 1{,}800$ and $t = o$.

$$\int_o^{T_r} \frac{dT}{\left(\frac{I}{.8}\right)^2 r - \frac{T}{(h+J)}} = \int_o^{1800} \frac{dt}{w}$$

$$T_r = (h + J)\left(\frac{I}{.8}\right)^2 r\left[1 - \cosh\frac{1800}{w(h+J)} + \sinh\frac{1800}{w(h+J)}\right]$$

In arriving at these formulæ two assumptions are made:

(*a*) That the resistance *r* is a constant—this is not strictly correct, as *r* is a function of T due to the temperature coefficient of the conductor.

(*b*) That the heat resistance J from armor to air is a constant. This is not strictly correct.

These discrepancies in a measure offset each other and are not of great importance with the low temperature rises under consideration. Of the above formulæ, I and III only are used in calculating wire sizes for use on Naval vessels, I for cables

carrying current continuously, and III for cables which will not carry maximum current for periods longer than thirty minutes at a time.

The following values for the constants used in making up the tables were determined by experimental work by the Navy Yard, New York, and by the New York Edison Co.:

$H = 116.\bar{5}$ $H = \Pi d J$ where d = diameter of cable overall.

$$J = \frac{H}{\Pi d}$$

K rubber = 285

K varnished cambric = 295

K reinforced rubber = 220

K lead to steel = 91

K lead = 1.23

From Bureau of Standards Circular 31, October 1, 1914, the resistance of copper = 10.371 ohms per mil foot at 20 degrees C., and 100 per cent conductivity. r = 1.0177 ohms/mil inch at 60 degrees C. and 98 per cent conductivity, and r = 1.0686 ohms/mil inch at 75 degrees C. and 98 per cent conductivity.

For determining values of w, the following densities were used:

Copper	.1142 grams per 1,000 circular mil inch.
Rubber	28.7 grams per cubic inch.
Lead	186.0 grams per cubic inch.
Steel	126.0 grams per cubic inch.
Varnished cambric	19.0 grams per cubic inch.
Reinforced rubber	21.4 grams per cubic inch.

The specific heats used were

Copper	= .393	Watt-seconds per gram per degrees C. rise.
Rubber	= .782	
Varnished cambric	= 1.672	
Lead	= .13	
Steel braid	= .494	

The value of w in formula III is made up as follows:

w_1 for copper = weight per inch $\times$ specific heat $\times$ 1
w_2 for insulation = weight per inch $\times$ specific heat $\times$ .85
w_3 for lead = weight per inch $\times$ specific heat $\times$ .70
w_4 for steel braid = weight per inch $\times$ specific heat $\times$ .65

$$w = w_1 + w_2 + w_3 + w_4$$

The factors 1, .85, .70 and .65 used in the above expressions indicate the temperature rise of the several materials per degrees C. rise in the copper.

The examples below are worked out to show the method of constructing the table.

414,020 CIRCULAR MIL RUBBER COVERED LEADED AND ARMORED CABLE CONTINUOUS CURRENT RATING.

r (ohms per inch at 60 degrees C.) $= \dfrac{1.0177}{414,020} = 2.458 \times 10^{-6}$

Conductor diameter ($2a$) .... = .742 inch

Diameter over rubber ($2b$) .... = 1.066

$\dfrac{2b}{2a}$ .... = 1.437

$\dfrac{K}{2\Pi} \log_e \dfrac{2b}{2a} = \dfrac{285}{2\Pi} \log_e 1.437$ .... = 16.43

Diameter over lead ($2c$) .... = 1.226

$\dfrac{2c}{2b}$ .... = 1.15

$\dfrac{K_1}{2\Pi} \log_e \dfrac{2c}{2b} = \dfrac{1.23}{2\Pi} \log_e 1.15$ .... = .027

Diameter overall ($2d$) .... = 1.396

$\dfrac{2d}{2c}$ .... = 1.138

$\dfrac{K_2}{2\Pi} \log_e \dfrac{2d}{2c} = \dfrac{91}{2\Pi} \log_e 1.138$ .... = 1.87

$h = 16.43 + .027 + 1.87$ .... = 18.33

$J = \frac{H}{\Pi d} = \frac{116.5}{\Pi \times 1.396} \ldots\ldots\ldots\ldots\ldots\ldots = 26.60$

$(h + J) \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots = 44.93$

$$I = .8 \sqrt{\frac{T}{r(h+J)}} = .8 \sqrt{\frac{15}{2.458 \times 10^{-6} \times 44.93}} = 294.5 \text{ amps.}$$

THIRTY-MINUTE RATING STARTING FROM ROOM TEMPERATURE.

$$I = .8 \sqrt{\frac{T_r}{r(h+J)\left[1 - \cosh \frac{1800}{w(h+J)} + \sinh \frac{1800}{w(h+J)}\right]}}$$

$r = 2.54 \times 10^{-6}$

$h + J = 44.93$ from above

$w_1 = .1142 \times 414.020 \times .393 \ldots\ldots\ldots\ldots\ldots = 18.55$

$w_2 = 28.7 \times \frac{\Pi}{4} (1.066^2 - .742^2) \times .782 \times .85 = 8.76$

$w_3 = 186 \times \frac{\Pi}{4} (1.226^2 - 1.066^2) \times .13 \times .70 = 4.87$

$w_4 = 126 \times \frac{\Pi}{4} (1.396^2 - 1.226^2) \times .494 \times .65 = 14.20$

$w = \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots 46.38$

$\frac{1{,}800}{w(h+J)} = \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots .867$

$$I = .8 \sqrt{\frac{25}{2.54 \times 10^{-6} \times 44.93 (1 - 1.40 + .98)}} = 493 \text{ amps.}$$

30,780 CIRCULAR MIL CABLE LEADED AND ARMORED VARNISHED CAMBRIC INSULATION.

CONTINUOUS CURRENT RATING.

r (ohms per inch at 75 degrees C.) ...... $= 34.71 \times 10^{-6}$

Conductor diameter ($2a$) .... = .201 inch

Diameter over tape ($2b$) .... = .383

$\frac{2b}{2a} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots = 1.91$

$\frac{K}{2\Pi} \log_e \frac{2b}{2a} = \frac{295}{2\Pi} \log_e 1.91 \ldots\ldots\ldots\ldots = 30.8$

Diameter over lead sheath ($2c$) .... = .453 inch

$$\frac{2c}{2b} \dots\dots\dots\dots\dots\dots\dots\dots\dots\dots = 1.185$$

$$\frac{K}{2\Pi} \log_e \frac{2c}{2b} = \frac{1.23}{2\Pi} \log_e 1.185 \dots\dots\dots\dots = .033$$

Diameter overall $(2d)$ .... $= .623$

$$\frac{2d}{2c} \dots\dots\dots\dots\dots\dots\dots\dots\dots\dots = 1.378$$

$$\frac{K}{2\Pi} \log_e \frac{2d}{2c} = \frac{91}{2\Pi} \log_e 1.378 \dots\dots\dots\dots = 4.63$$

$$h = 30.8 + .033 + 4.63 \dots\dots\dots\dots\dots = 35.46$$

$$J = \frac{H}{\Pi d} = \frac{116.5}{\Pi \times .623} \dots\dots\dots\dots\dots = 59.50$$

$$h + J \dots\dots\dots\dots\dots\dots\dots\dots\dots = 94.96$$

$$I = .8\sqrt{\frac{30}{34.71 \times 10^{-6} \times 94.96}} = 76.5 \text{ ampères.}$$

THIRTY-MINUTE RATING.

$$r = 34.71 \times 10^{6}$$

$$h + J = 94.96$$

$$w_1 = .1142 \times 30.780 \times .393 \dots\dots\dots\dots\dots = 1.382$$

$$w_2 = 19 \times \frac{\Pi}{4}(.383^2 - .201^2) \times 1.672 \times .85 = 2.250$$

$$w_3 = 186 \times \frac{\Pi}{4}(.453^2 - .383^2) \times .13 \times .70 = .785$$

$$w_4 = 126 \times \frac{\Pi}{4}(.623^2 - .453^2) \times .494 \times .65 = 5.810$$

$$w = w_1 + w_2 + w_3 + w_4 \dots\dots\dots\dots\dots = 10.227$$

$$I = .8\sqrt{\frac{30}{34.71 \times 10^{-6} \times 94.96\left[1 - \cosh\frac{1800}{10.23 \times 94.96} + \sinh\frac{1800}{10.23 \times 94.96}\right]}} = 83.2 \text{ ampères.}$$

The values worked out in the examples above agree very closely with those given in the table. In making up the tables the various currents were worked out and the results plotted on logarithmic paper, using as coordinates ampères versus size of conductor in circular mils. The values for a particular type of cable will lie practically on a straight line. This is quite true for the continuous current values where the relation is practically a simple variation of current with circular mil area.

For the intermittent ratings the relation between circular mil area and current-carrying capacity is not quite a straight line, due to the greater relative thermal capacity of the copper, armor and insulating materials of the larger-sized cable. This would tend to make the curve depart upward from the straight line.

TABLES SHOWING MAXIMUM ALLOWABLE LOAD IN AMPERES FOR CONTINUOUS AND INTERMITTENT LOADS ON NAVY STANDARD CABLES.

RUBBER INSULATION CABLES - LEADED AND ARMORED

Circular Mils		Continuous Load		Intermittent Load	
Approximate	Actual	Single	Duplex	Single	Duplex
4.000	4,494	18	15	23	20
9,000	9,030	26	22	36	31
11,000	11,340	31	26	41	35
14,000	14,350	35	30	47	40
18,000	18,060	41	35	55	47
23,000	22,820	47	40	64	54
30,000	30,780	56	48	77	65
40,000	38,950	66	56	88	75
50,000	49,020	75	64	102	87
60,000	59,940	85	72	117	99
75,000	75,850	99		141	
100,000	98,820	118		172	
125,000	125,050	135		205	
150,000	157,380	158		240	
200,000	198,860	184		283	
250,000	250,710	214		347	
300,000	296,660	238		382	
375,000	374,010	280		455	
400,000	414,020	295		495	
500,000	521,970	344		585	
650,000	657,860	402		715	
800,000	829,310	475		835	

TABLES SHOWING MAXIMUM ALLOWABLE LOAD IN AMPERES FOR CONTINUOUS AND INTERMITTENT LOADS ON NAVY STANDARD CABLES.

RUBBER INSULATION CABLES - ARMORED.

Circular Mils		Continuous Load		Intermittent Load	
Approximate	Actual	Single	Duplex	Single	Duplex
4,000	4,494	17	14	19	16
9,000	9,030	24	20	30	26
11,000	11,340	28	24	34	29
14,000	14,350	32	27	39	33
18,000	18,060	38	32	46	39
23,000	22,820	45	37	54	46
30,000	30,780	52	44	65	55
40,000	38,950	61	52	74	63
50,000	49,020	69	59	86	73
60,000	59,940	78	66	99	84
75,000	75,850	92		119	
100,000	98,820	110		146	
125,000	125,050	127		175	
150,000	157,380	149		206	
200,000	198,860	175		246	
250,000	250,710	204		303	
300,000	296,660	226		336	
375,000	374,010	266		403	
400,000	414,020	280		437	
500,000	521,970	327		520	
650,000	657,860	383		662	
800,000	829,310	451		785	

TABLES SHOWING MAXIMUM ALLOWABLE LOAD IN AMPERES FOR CONTINUOUS AND INTERMITTENT LOADS ON NAVY STANDARD CABLES.

RUBBER INSULATION CABLES - PLAIN BRAIDED.

Circular Mils		Continuous Load		Intermittent Load	
Approximate	Actual	Single	Duplex	Single	Duplex
4,000	4,494	16	14	18	15
9,000	9,030	22	19	28	24
11,000	11,340	27	23	32	27
14,000	14,350	30	26	37	31
18,000	18,060	36	31	43	37
23,000	22,820	44	37	50	43
30,000	30,780	49	42	61	52
40,000	38,950	58	49	70	60
50,000	49,020	66	56	81	69
60,000	59,940	70	60	93	79
75,000	75,850	87		113	
100,000	98,820	106		138	
125,000	125,050	120		165	
150,000	157,380	142		195	
200,000	198,860	167		232	
250,000	250,710	195		286	
300,000	296,660	218		317	
375,000	374,010	256		362	
400,000	414,020	270		420	
500,000	521,970	316		503	
650,000	657,860	373		654	
800,000	829,310	441		767	

TABLES SHOWING MAXIMUM ALLOWABLE LOAD IN AMPERES FOR CONTINUOUS AND INTERMITTENT LOADS ON NAVY STANDARD CABLES.

VARNISHED CAMBRIC CABLES - LEADED AND ARMORED.

Circular Mils		Continuous Load		Intermittent Load	
Approximate	Actual	Single	Duplex	Single	Duplex
4,000	4,494	24	20	25	21
9,000	9,030	36	31	38	32
11,000	11,340	42	36	45	38
14,000	14,350	48	41	52	44
18,000	18,060	56	48	61	52
23,000	22,820	64	54	70	60
30,000	30,780	77	65	86	73
40,000	38,950	90	77	100	85
50,000	49,020	104	88	117	99
60,000	59,940	117	99	134	114
75,000	75,850	136		158	
100,000	98,820	162		189	
125,000	125,050	187		223	
150,000	157,380	218		266	
200,000	198,860	252		316	
250,000	250,710	294		380	
300,000	296,660	326		432	
375,000	374,010	382		515	
400,000	414,020	409		560	
500,000	521,970	474		668	
650,000	657,860	555		790	
800,000	829,310	650		950	

TABLES SHOWING MAXIMUM ALLOWABLE LOAD IN AMPERES FOR CONTINUOUS AND INTERMITTENT LOADS ON NAVY STANDARD CABLES.

VARNISHED CAMBRIC CABLES - ARMORED.

Circular Mils		Continuous Load		Intermittent Load	
Approximate	Actual	Single	Duplex	Single	Duplex
4,000	4,494	23	20	23	20
9,000	9,030	33	28	34	29
11,000	11,340	39	33	40	34
14,000	14,350	45	38	46	39
18,000	18,060	52	44	54	46
23,000	22,820	60	51	62	53
30,000	30,780	72	61	77	65
40,000	38,950	84	71	89	76
50,000	49,020	97	82	104	88
60,000	59,940	109	93	120	102
75,000	75,850	127		142	
100,000	98,820	151		170	
125,000	125,050	174		202	
150,000	157,380	204		242	
200,000	198,860	237		291	
250,000	250,710	278		350	
300,000	296,660	308		398	
375,000	374,010	361		474	
400,000	414,020	388		515	
500,000	521,970	450		615	
650,000	657,860	530		736	
800,000	829,310	625		905	

TABLES SHOWING MAXIMUM ALLOWABLE LOAD IN AMPERES FOR CONTINUOUS AND INTERMITTENT LOADS ON NAVY STANDARD CABLES.

VARNISHED CAMBRIC CABLES - PLAIN BRAIDED.

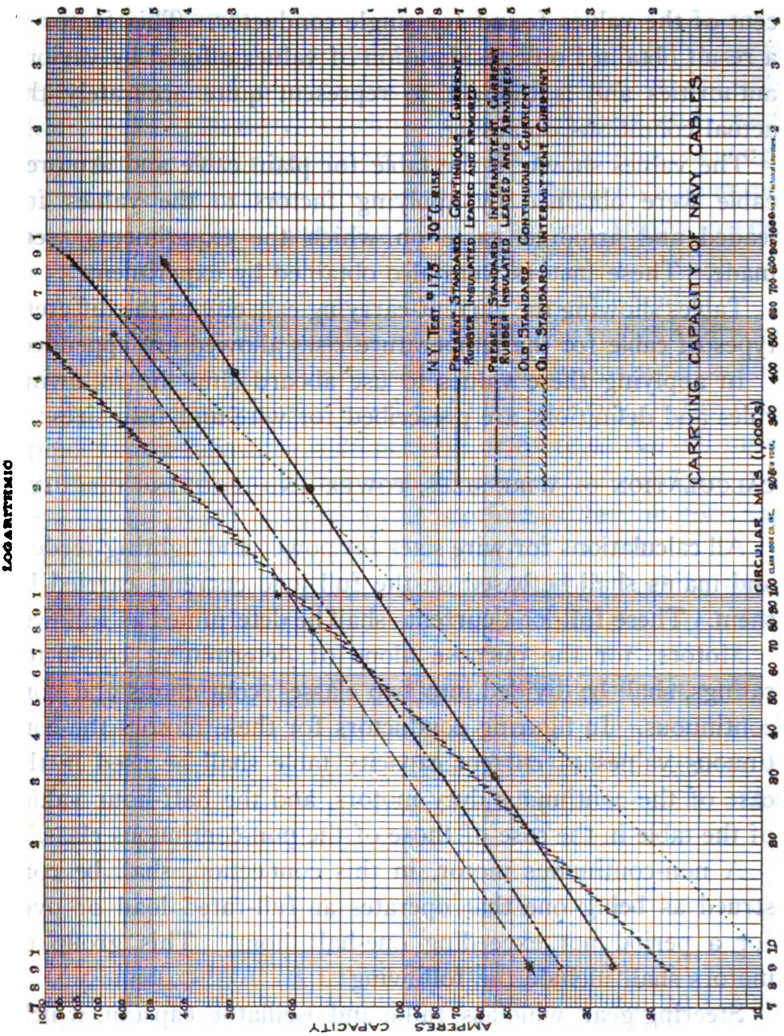
Circular Mils		Continuous Load		Intermittent Load	
Approximate	Actual	Single	Duplex	Single	Duplex
4,000	4,494	21	18	21	18
9,000	9,030	31	26	31	26
11,000	11,340	36	31	36	31
14,000	14,350	42	36	42	36
18,000	18,060	49	42	49	42
23,000	22,820	56	48	57	49
30,000	30,780	67	57	70	60
40,000	38,950	79	67	81	69
50,000	49,020	91	77	95	81
60,000	59,940	96	82	110	94
75,000	75,850	113		130	
100,000	98,820	144		158	
125,000	125,050	166		188	
150,000	157,380	196		228	
200,000	198,860	230		274	
250,000	250,710	268		330	
300,000	296,660	298		378	
375,000	374,010	350		450	
400,000	414,020	375		490	
500,000	521,970	435		588	
650,000	657,860	515		705	
800,000	829,310	605		865	

TABLES SHOWING MAXIMUM ALLOWABLE LOAD IN AMPERES FOR CONTINUOUS AND INTERMITTENT LOADS ON NAVY STANDARD CABLES.

VARNISHED CAMBRIC WITH REINFORCED RUBBER CABLES.

Circular Mils		Armored		Without Armor	
Approximate	Actual	Continuous Single	Intermittent Single	Continuous Single	Intermittent Single
250,000	250,710	280	376	272	357
300,000	296,660	312	429	302	409
375,000	374,010	362	512	352	490
400,000	414,020	387	552	376	528
500,000	521,970	449	660	436	634
650,000	657,860	525	790	510	757
800,000	829,310	619	940	610	902

These relations are shown on the curves attached.



The table in its final form was picked off from the faired curve, which explains the small discrepancies noted in the examples worked out.

The table for twin conductor was made up by taking 85 per cent of the values found for single conductor. This factor is a result obtained on consideration of values offered by various authorities and is thought to represent quite accurately the actual conditions obtaining.

The values shown in the table for plain cable and armored cable were obtained by applying factors to the values for leaded and armored cable on which the experiments were made. These factors were also obtained by experiment.

Tables showing the current-carrying capacity of the different types of cable for the temperature rises allowed are appended.

In applying these values to use aboard ship the following rules and definitions are prescribed for wire and fuse sizes:

CALCULATION OF WIRE SIZES, POWER AND LIGHTING CIRCUITS.

All calculations for wire sizes for power and lighting feeders and mains shall be based on the full load currents carried by them. These full-load currents shall be determined as follows:

Motors, for the purpose alone of determining conductor ratings, shall be divided into two classes, continuous and non-continuous. In figuring conductors for these motors the continuous values of current capacity table shall be used in the case of the continuous duty motors, and the half-hour values of the table in the case of those of the non-continuous class.

A non-continuous motor, in this connection, shall be construed as being one that operates at full-rated load or over for a period not exceeding one-half hour. This group of motors shall include the following:

Steering gear, windlass, radio and oscillator, capstan, winch, turret machinery, boat crane, turbine and shaft turning, torpedo hoist, and sounding machine motors.

Continuous motors shall include in general those not plainly of the non-continuous class and, specifically, those which operate continuously at times at full-rated load for periods of over one-half hour. A list of such motors and apparatus is given for guidance:

Ventilation, blower, portable fan, ammunition hoist, conveyor, pump, engine-room auxiliary, air compressor, ice machine, galley, laundry, machine shop, workshop and print shop motors; motor-generators for battery charging, welding, interior communication, fire control and submersible pumping apparatus, etc.; and apparatus such as heaters, stoves, ovens, searchlights.

Auxiliary power shall generally be rated continuous.

In power feeders or mains when both continuous and non-continuous duty motors are used, the continuous rated cables shall be used in case the rated full-load capacity of the continuous duty apparatus is 25 per cent or more of the full-rated load of the circuit. If less than 25 per cent of full-load current of the circuit is made up of continuous duty motors, the half-hour rating of cables shall be used.

The rated ampère capacity of cable selected shall be equal to, or in case of excessive drop, greater than the rated full-load current of the circuit.

In selection of all cables, the drop in voltage shall be kept within limits of existing specifications.

In the following paragraphs the rated load of a motor or other piece of electrical apparatus shall be the ampères called for on manufacturer's nameplate.

FULL-LOAD CURRENT.

Full-load current of circuit between generator and switchboard shall be the full-rated load of generator plus 33 1/3 per cent, and between generator board and distribution board shall be the full-load rating of the several ship's service gen-

erators in one engine room, omitting exciter, if any, plus 33 1/3 per cent.

The combined *full-load current* of power and lighting-bus tie feeders between distribution boards in cases where lighting and power cables constitute independent circuits shall be equal to rated capacity of ship's service generators in one dynamo or engine room.

The *full-load current* of bus tie cables between distribution boards when these cables carry jointly the lighting and power loads shall be that equivalent to one-half of the overload capacity (133 per cent) of ship's service generators in one engine or dynamo room.

Bus tie feeders between generator boards and distribution boards and between forward and after distribution boards, and also generator cables, shall be figures so as to observe the following conditions:

(*a*) Drop in voltage on power circuits from generator board to motor feeding from its own distribution board shall not exceed 5 per cent, and in case of motor feeding from opposite generator board, 6 per cent.

(*b*) Wiring for lighting systems will be designed on the basis of a maximum allowable drop of 2½ per cent, calculated from adjacent generator switchboard, and 3 per cent from the distant generator switchboard, based on calculated full-load currents hereinafter defined.

(*c*) Maximum allowable drop, generator to distribution board, 1 per cent.

(*d*) Allowable drop, generator board to adjacent distribution board, one-half of 1 per cent.

(*e*) The allowable drop in voltage over such feeders between distribution boards as shall be utilized for lighting service shall not exceed one-half of 1 per cent, based on a load equivalent to 1/3 of the sum of the *full-load currents* of all lighting feeders of forward or after lighting systems. This shall be considered the actual lighting load in the following paragraph:

(*f*) The allowable drop in voltage due to power load over the remainder of these feeders between distribution boards shall not exceed 1 per cent, a load current being assumed equivalent to the total generator load described above less the actual lighting load described in (*e*).

(*g*) In cases where one set of bus tie feeders between distribution boards carries jointly lighting and power load, the drop in voltage, under *full-load current* condition specified for this arrangement in paragraph above, shall not exceed 1 per cent.

(*h*) The total cross-section of all conductors between generator board and distribution board shall be at least equal to the total cross-section of cables between generator and generator board.

(*i*) In cases where the distribution panels and the generator panels are integral parts of the same switchboards the drop in voltage between generators and generator panels shall not exceed one-half of 1 per cent.

The *full-load current* of a feeder or main running to but one motor shall be the rated load of the motor.

The *full-load current* of a feeder or main running to a heating device or heating devices shall be the full-load current of the heating device or the sum of the full-load currents of the several heating devices.

The *full-load current* of a main running to several motors and heating devices shall be the sum of the full-load rating of the motors and heating devices plus the sum of the current capacities of the additional outlets on the main, if any.

The *full-load current* of a feeder from distribution board to an auxiliary power panel shall be the sum of the full-load currents of the mains running from the panel plus the sum of the current capacities of spare outlets or switches on the panel.

The *full-load current* of a feeder running direct to junction boxes supplying several motors and heating devices with no

auxiliary panel shall be the sum of the full-load currents of the motors and heating devices.

The *full-load current* of a lighting feeder running direct to a lighting panel or to feeder junction boxes will be equal to the sum of the full-load currents of the lighting mains feeding from the panel or junction boxes.

The *full-load current* of a lighting main running from a lighting panel or feeder junction box will be equal to the sum of the full-load currents of all sub-mains and branches feeding from it.

The *full-load current* of a lighting sub-main will be equal to the sum of the full-load currents of all branches feeding from the distribution box or junction box.

Lighting feeders to lighting panels or to feeder junction boxes will in no case be less than 9,030 c.m. twin conductor cable.

Lighting cables will in no case be less than 4,494 c.m. twin conductor, except in special cases such as signal, running and anchor light systems. The lighting sub-mains to distribution boxes should be of such cross-section that the drops in them will not cause undue increases in sizes of feeders and mains.

FUSE SIZES.

The circuits included in the accompanying rules will comprise feeders, mains and sub-mains. The following definitions are given as applying in this instance:

(*a*) A feeder shall be considered as a circuit fed directly from main power or lighting distribution board.

(*b*) A main shall be considered as a circuit receiving its energy directly from a feeder.

(*c*) A sub-main shall be considered as a circuit feeding directly from a main and supplying one or more distribution boxes.

The rated capacity of fuse for a power feeder from distribution board shall be equal to the full-load current of the

circuit plus 50 per cent of the current of largest motor, in no case, however, to be less than 25 per cent in excess of total rated ampères of connected load.

The rated capacity of fuse for lighting feeders from lighting distribution board shall be equal to the full-load current of the circuit plus 25 per cent.

The rated capacity of fuse on power mains and sub-mains shall be equal to the full-load current of such mains or sub-mains plus 50 per cent of the current of largest motor, in no case, however, to be less than 25 per cent in excess of total rated ampères of connected load.

The rated capacity of fuse on lighting mains or sub-mains shall be equal to the full-load current of such mains or sub-mains.

The rated capacity of power mains or sub-mains carrying heaters alone shall be equal to the combined full-load current of the connected heaters.

With the table of current-carrying capacities given and the rules for applying these tables, it is believed that no copper will be wasted in laying out the various systems aboard ship while there will be sufficient copper provided to carry the desired currents without overheating.

U. S. S. *EAGLE* BOATS.

DESCRIPTION.

By Commander Carlos Bean, U. S. Navy, Member.

(Plates Nos. 1 to 6 show construction of hull and launching devices.)

In February, 1918, the Ford Motor Company, of Detroit, Michigan, contracted to build one hundred 200-foot patrol boats for the U. S. Navy, on a cost plus fixed profit basis. The number of boats originally contracted for was reduced to sixty after the armistice was signed.

The site selected for the shipyard lies to the west of the city limits, about eight miles from the center of the city, near the River Rouge.

Active work in erecting buildings and dredging a slip communicating with the River Rouge was begun in March, 1918. This slip is about one-half a mile long, three hundred feet wide, and twenty-eight feet deep. The cost of the necessary buildings and equipment was about three and one-half million dollars.

The principal buildings erected were: The fabricating shop, the assembly building and store building. The transfer table, the hydraulic launching table and fit-out docks, constructed along with the above buildings, complete the plant equipment.

Plates, frames, shapes, etc., were manufactured in the fabricating shop, which connects with the assembly building.

The assembly building is 1,750 feet long and 300 feet wide. In it were laid three main tracks for specially made heavy trucks on which the hulls of the *Eagle* boats were built. As work progressed on a hull, it was moved on its trucks from one assembly stage to the next until it had passed through seven stages, when it was ready for launching. Each one of the three main tracks would accommodate a line of seven

hulls, making a total of twenty-one hulls under continuous assembly.

The transfer table is simply a truck about 190 feet long and 30 feet wide, which runs on heavy tracks parallel to the end of the assembly building. Motive power is furnished by a 30 horsepower motor.

When a hull has passed through its last stage of assembly in the assembly building, it is hauled on its own trucks out on to the transfer table, on which it is transported to a point opposite the hydraulic launching table. The hull, still on its own trucks, is then hauled on to the hydraulic launching table, lowered into the water, floats off its ways when water borne, and is then towed clear. The launching table is then raised, the trucks placed on the transfer table, and returned to the first stage in the assembly building.

The launching table is an hydraulic lift 190 feet in length and operated by eight hydraulic engines, four on each side. The capacity of the table is about 600 tons. The total time required to launch a hull, starting from the last stage in the assembly building, is about 40 minutes.

The *Eagle* boat is a single-screw vessel, driven by a high-speed Poole turbine through a planetary reduction gear, and is designed for a speed of 18 knots at about 500 tons trial displacement with the engine developing about 2,000 shaft horsepower.

GENERAL DIMENSIONS.

Length between perpendiculars, feet	200
Length over all, feet and inches	200–9
Beam molded at mid perpendicular, feet and inches	25–6
Depth molded from stem to bulkhead No. 59, feet and inches	18-6
Depth molded at after perpendicular, feet and inches	14–6
Mean trial displacement, tons (about)	500
Mean draft to bottom of keel at mean trial displacement, feet and inches	7–3
Fuel oil, approximate full load capacity, tons	90
Fuel oil, approximate emergency capacity, tons	145
Frame spacing, inches	21

Fuel oil is carried in six "D" compartments and three "A" compartments. Fresh water for feed and ship's use is carried in three tanks between bulkhead No. 59 and No. 65 in "C" compartment.

Amount of feed water carried, gallons	5,680
Amount of ship's water carried, gallons	2,400

COMPLEMENT.

Commanding officer	1
Wardroom officers	3
Crew, including chief petty officers	54
	58

BATTERY.

Two four-inch 50 caliber rapid fire rifles.

One three-inch 50 caliber anti-aircraft rifle, high power, flat trajectory type.

One Y gun, depth charge projector.

Two machine guns.

Depth charge release gear is intalled on port and starboard sides of after part of main deck.

SMALL BOATS CARRIED.

One 21-foot motor dory.

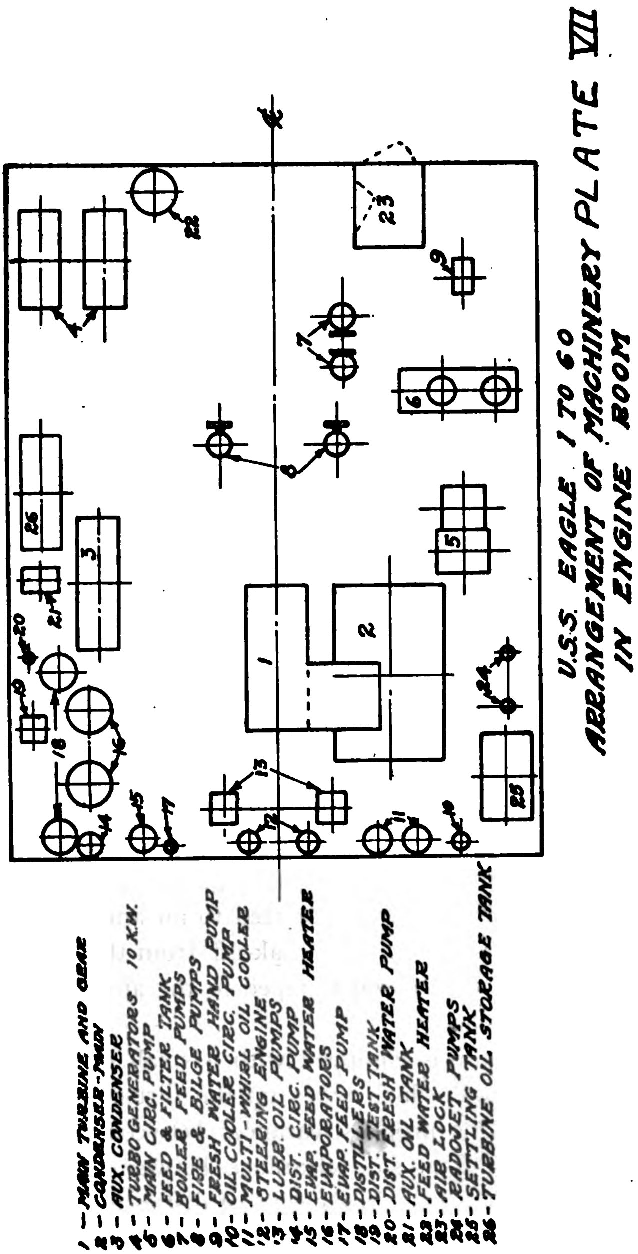
One 24-foot whale boat.

The boats are stowed on heavy curved channel davits. The davits rig in and out by means of a sheathed screw gear operated by a crank.

MACHINERY.

(See Plate VII for arrangement of machinery space.)

The main engine is a Poole high-speed turbine of the compound impulse velocity stage type, with a working pressure of 235 pounds. It is designed to develop 2,000 shaft horsepower at 4,000 revolutions per minute and is geared down in the ratio of 8.4 to 1.


1 — MAIN TURBINE AND GEAR
2 — CONDENSER-MAIN
3 — AUX. CONDENSER
4 — TURBO GENERATORS 10 K.W.
5 — MAIN CIRC. PUMP
6 — FEED & FILTER TANK
7 — BOILER FEED PUMPS
8 — FIRE & BILGE PUMPS
9 — FRESH WATER HAND PUMP
10 — OIL COOLER CIRC. PUMP
11 — MULTI-WHIRL OIL COOLER
12 — STEERING ENGINE
13 — LUBR. OIL PUMPS
14 — DIST. CIRC. PUMP
15 — EVAP. FEED WATER HEATER
16 — EVAPORATORS
17 — EVAP. FEED PUMP
18 — DISTILLERS
19 — DIST. TEST TANK
20 — DIST. FRESH WATER PUMP
21 — AUX. OIL TANK
22 — FEED WATER HEATER
23 — AIR LOCK
24 — RADOJET PUMPS
25 — SETTLING TANK
26 — TURBINE OIL STORAGE TANK
U.S.S. EAGLE 1 TO 60
ARRANGEMENT OF MACHINERY IN ENGINE ROOM
PLATE VII

There are six pressure stages in the ahead turbine—the first or high-pressure stage is velocity compounded. Incorporated in the same casing and at the after end is the backing turbine, which consists of one pressure stage, velocity compounded. It has about 30 per cent of the power of the ahead turbine.

(For details of rotor see Plate VIII.)

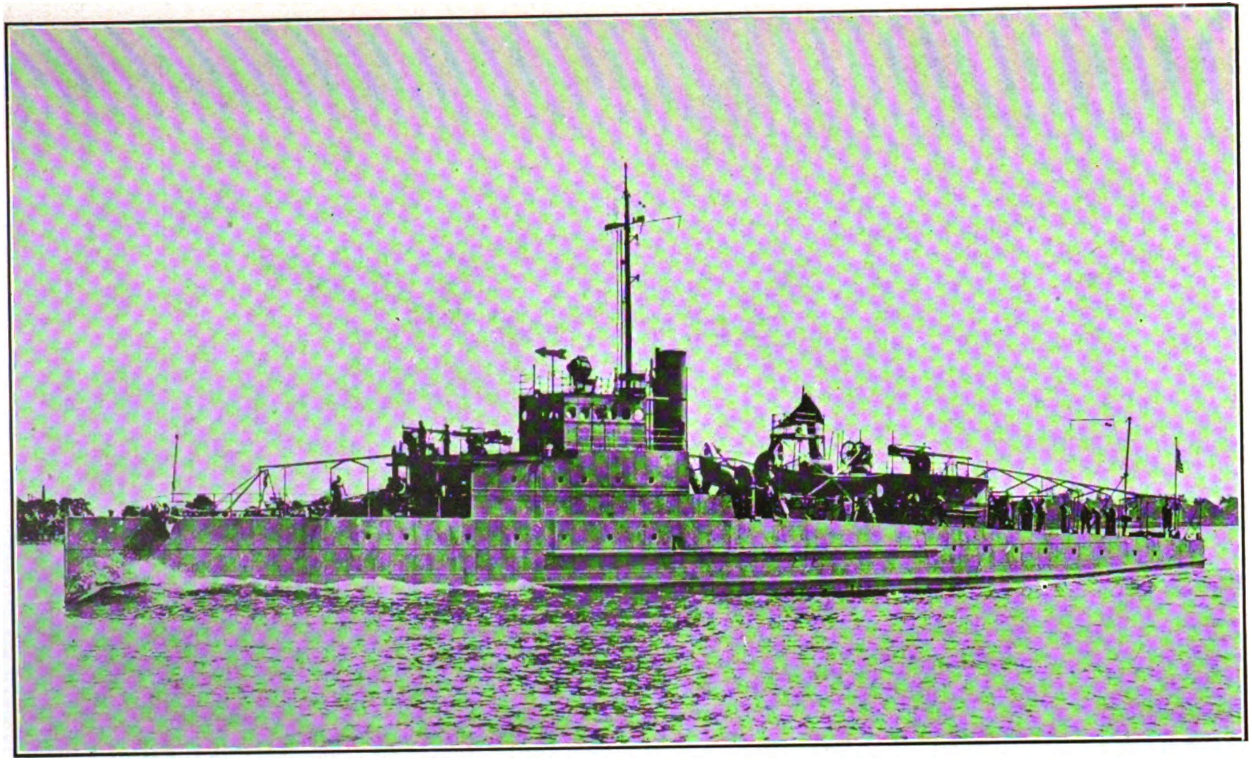
The rotor wheels are assembled in the central part of the shaft, keyed in place, distance pieces being placed between the wheels to maintain the positions. A nut on each end of the central portion of the shaft is screwed hard against the first and last wheels. These securing nuts are locked in place by set screws.

The first rotor wheel in the ahead turbine and the astern rotor wheel are of the same size and construction. Each carry two sets of blades.

The rotor shaft revolves in self-aligning bearings of the spherical seat type. The forward bearing is located in the forward end of the turbine casing, and the after bearing in the forward end of the reduction gear casing.

The central part of the shaft is 27 inches long and 6.998 inches in diameter. The shaft decreases in size on both sides of this central part to 5½ inches in diameter for lengths of 9¾ inches. These parts of the shaft provide smooth cylindrical surfaces, which revolve in a series of radial slotted projections forming the shaft gland packing. Sealing steam from the auxiliary exhaust line is admitted to an annular space at the center of the packing. The leak-off from the shaft packing at the high-pressure end is piped to the annular space in the low-pressure packing.

The turbine rotor is equipped with a small thrust bearing at the forward end, which takes up the thrust of the rotor shaft and maintains proper clearances between moving and stationary parts. The side clearance between moving and stationary parts is adjusted by means of distance pieces behind the thrust blocks. The after end of the rotor shaft is tapered



"EAGLE" No. 57.

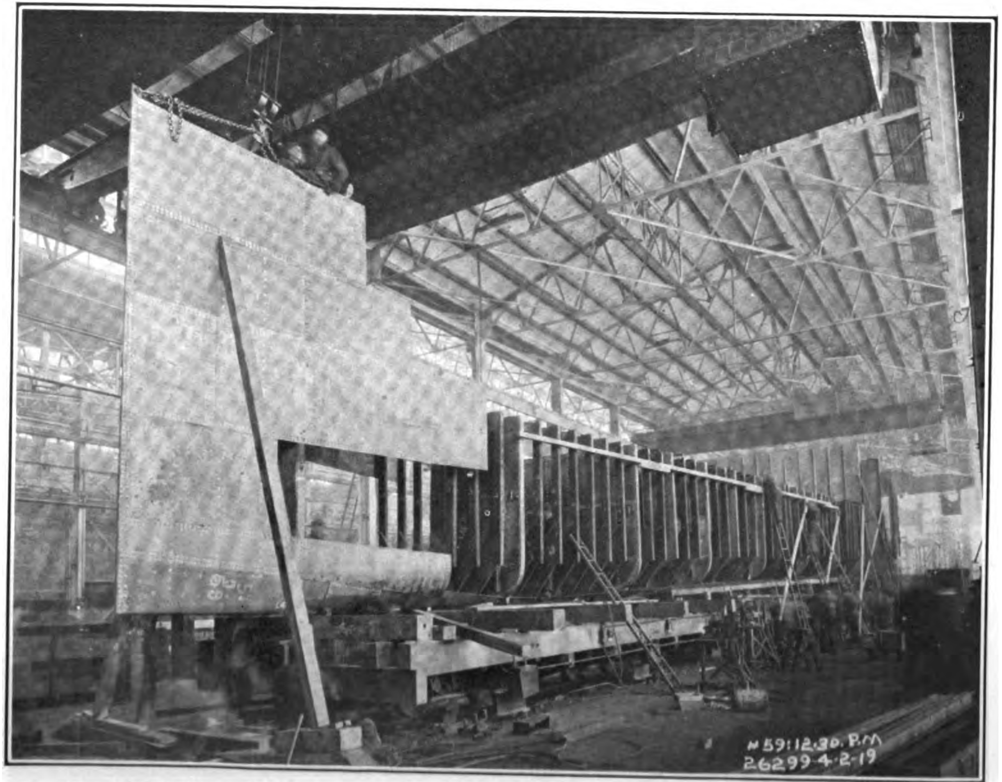


PLATE I.—HULL UNDER CONSTRUCTION IN ASSEMBLY BUILDING, SHOWING TRUCKS.

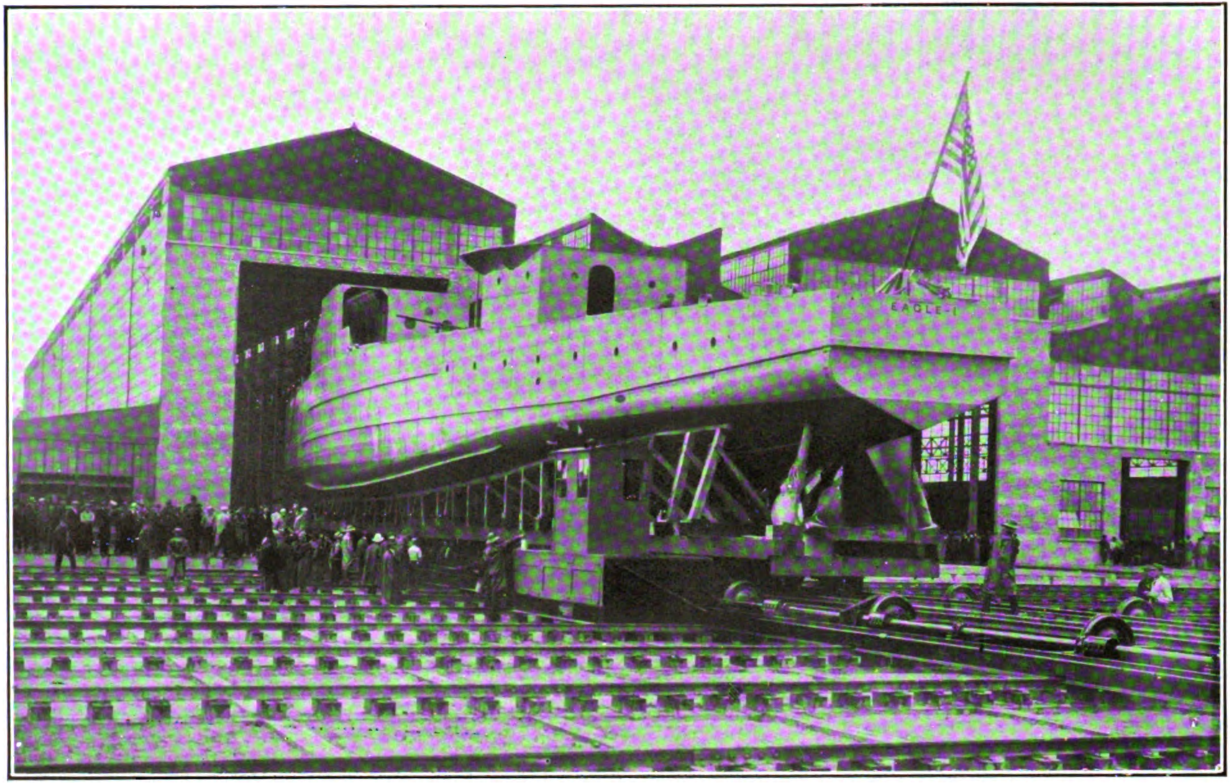


Plate II.—Hull Leaving Assembly Building and Being Hauled Onto Transfer Table.

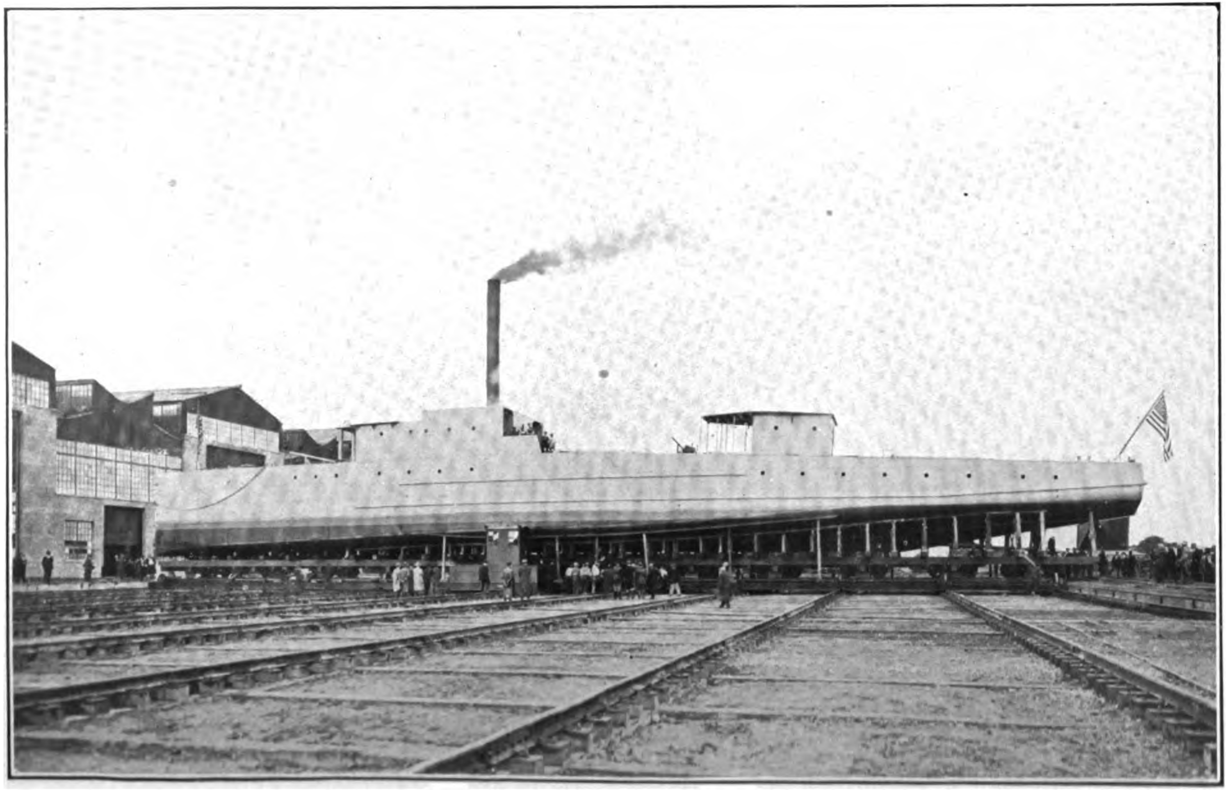


PLATE III.—HULL ON TRANSFER TABLE.

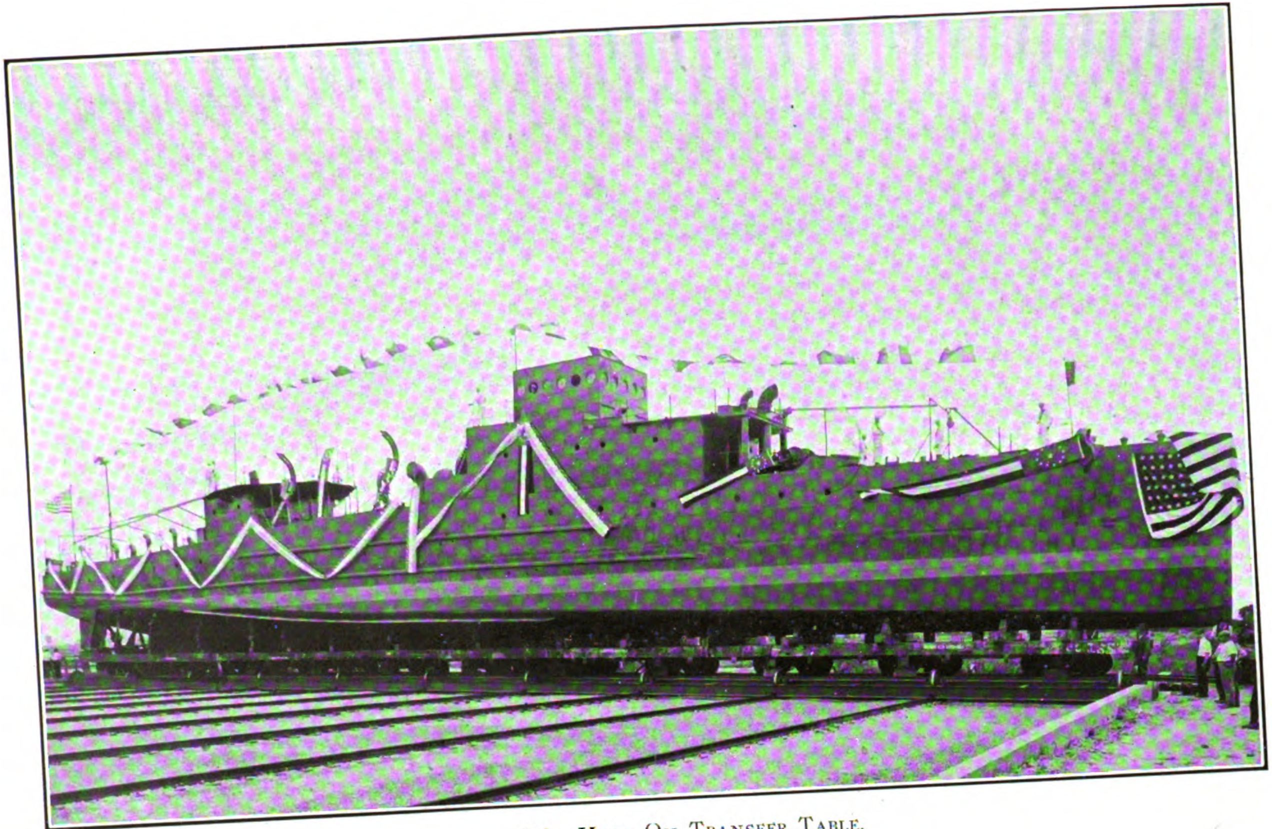


Plate IV.—Hull On Transfer Table.

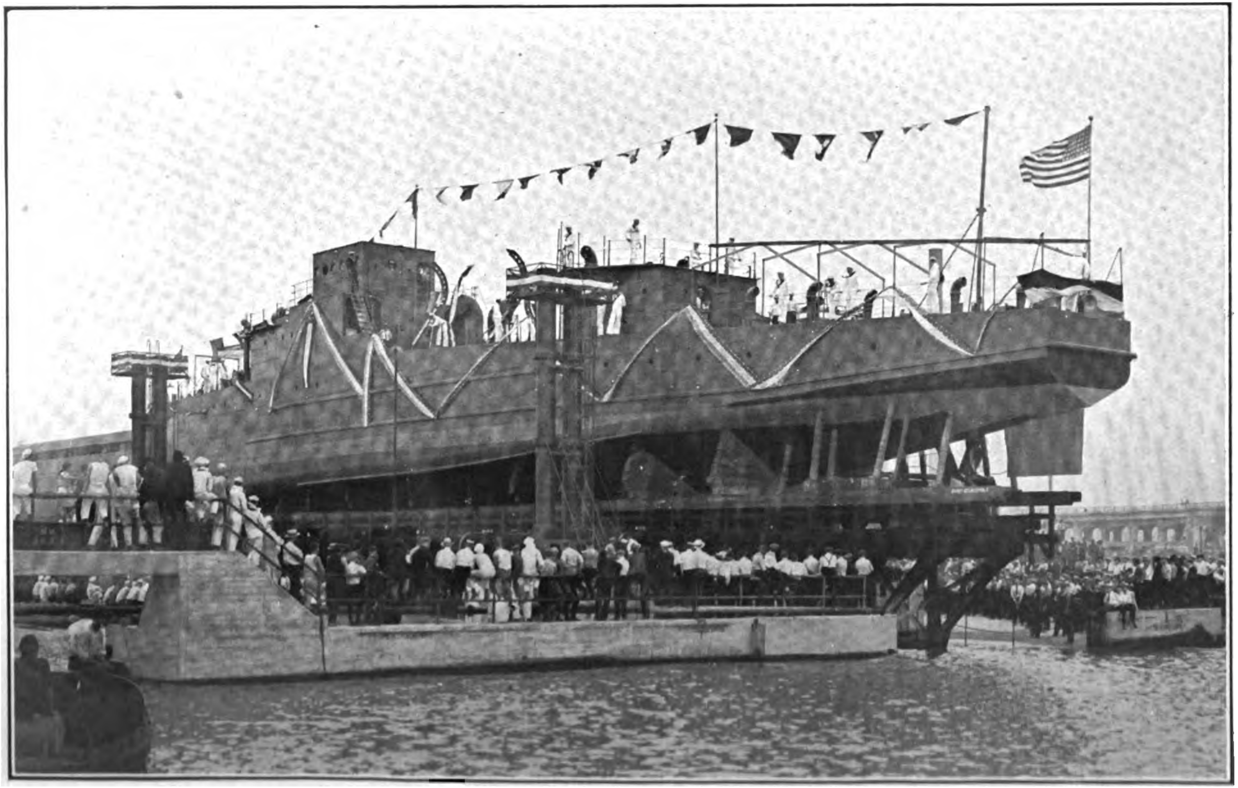


PLATE V.—HULL ON LAUNCHING TABLE.

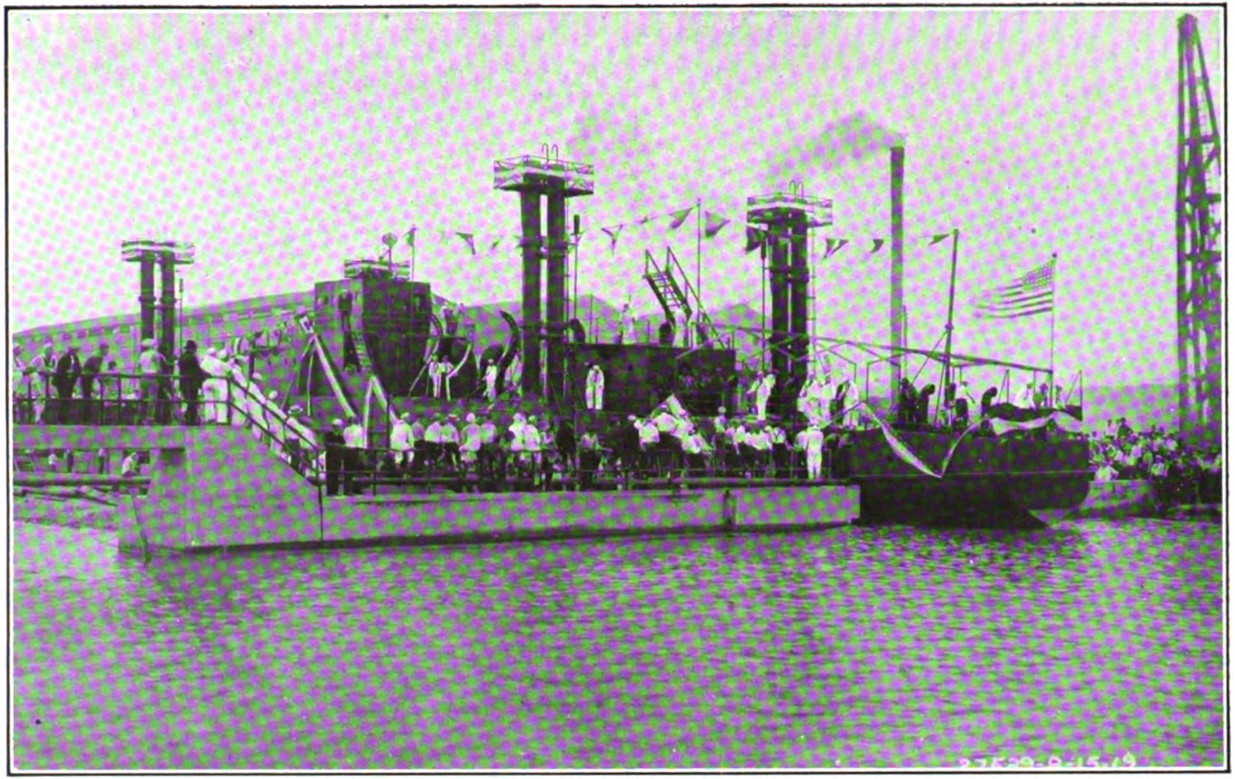


PLATE VI.—HULL FLOATING OFF LAUNCHING TABLE.

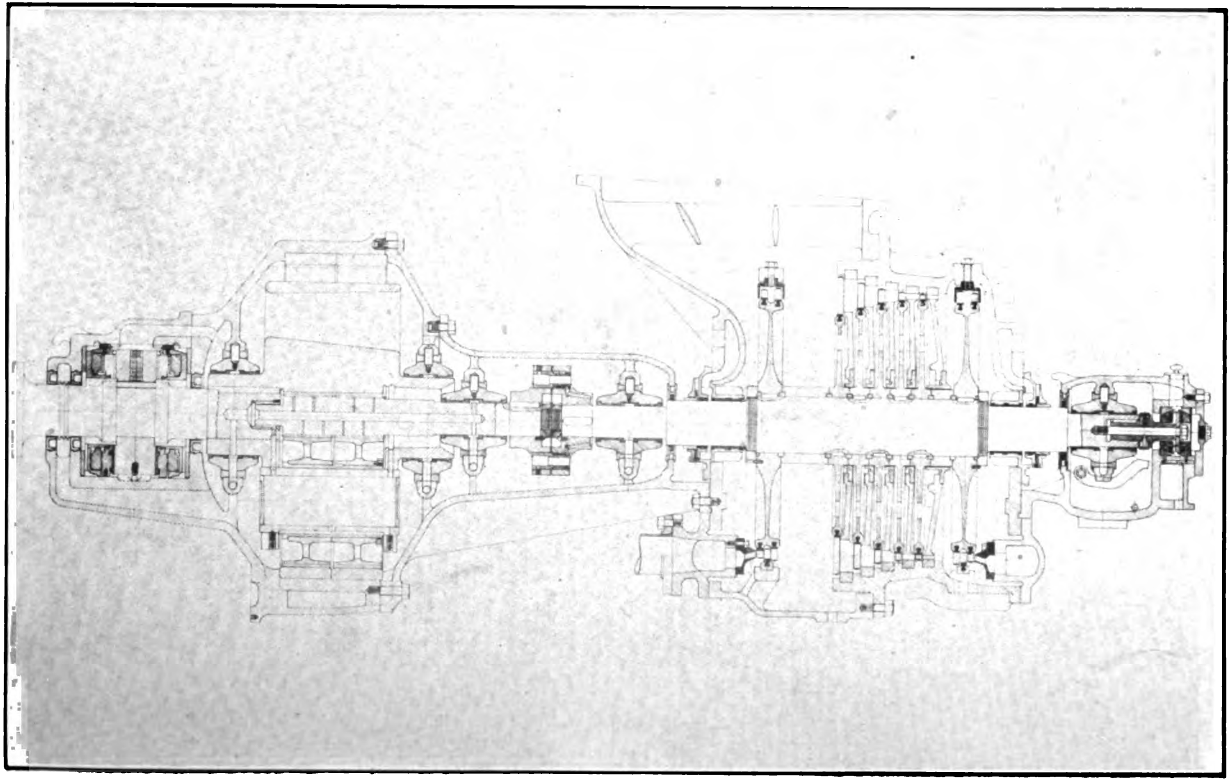


Plate VIII.—Rotor.

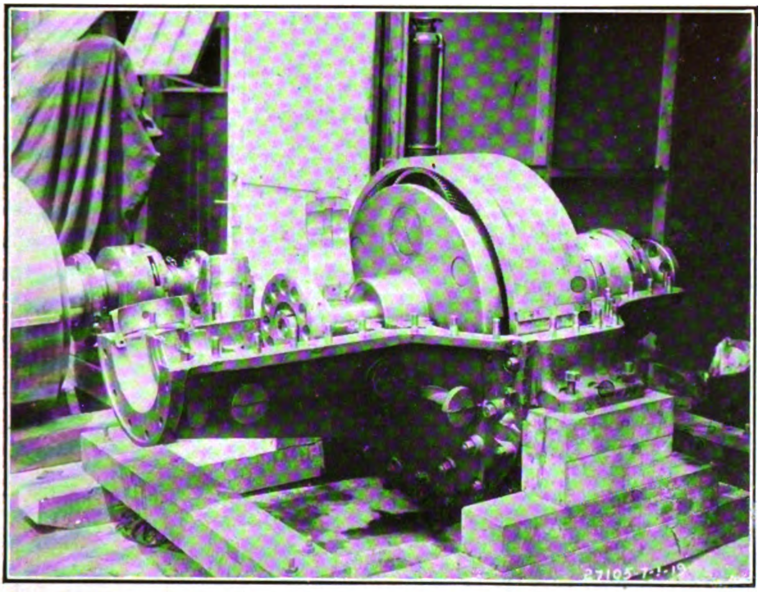


Plate X.—Reduction Gear.

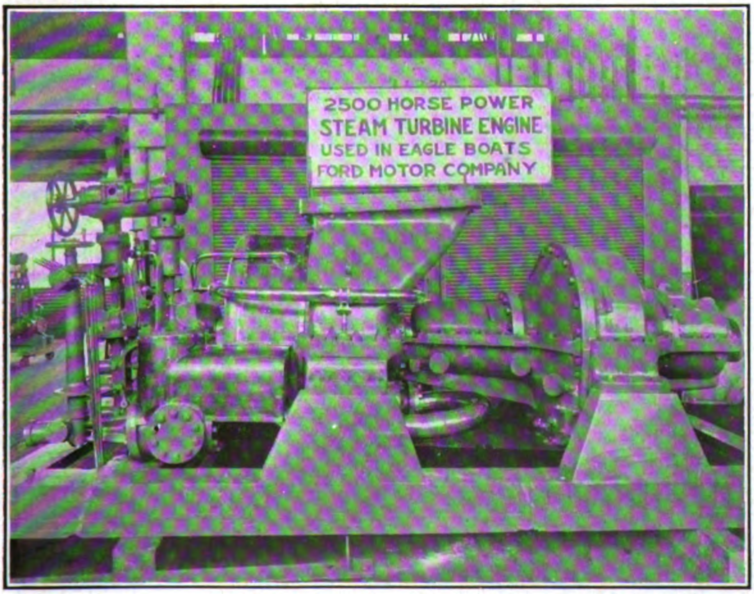


Plate XI.—Main Engine Assembly.

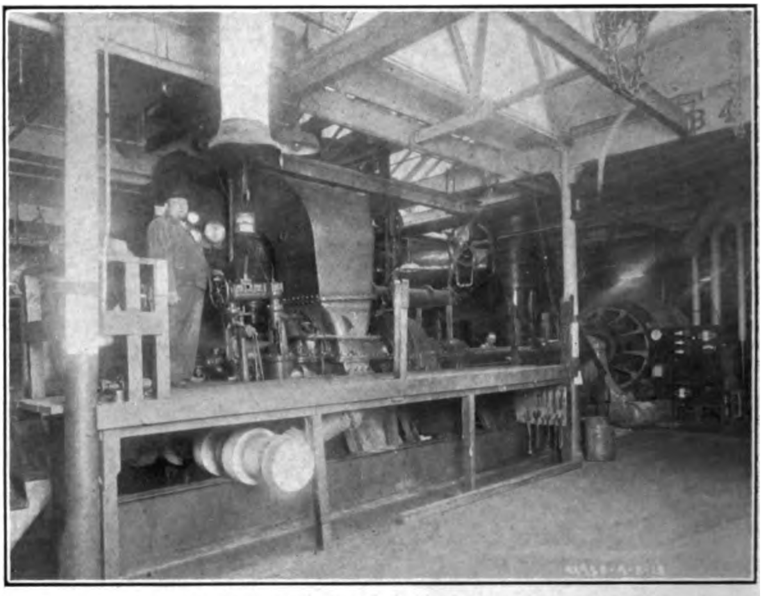


PLATE XII.—TURBINE TEST BLOCK.

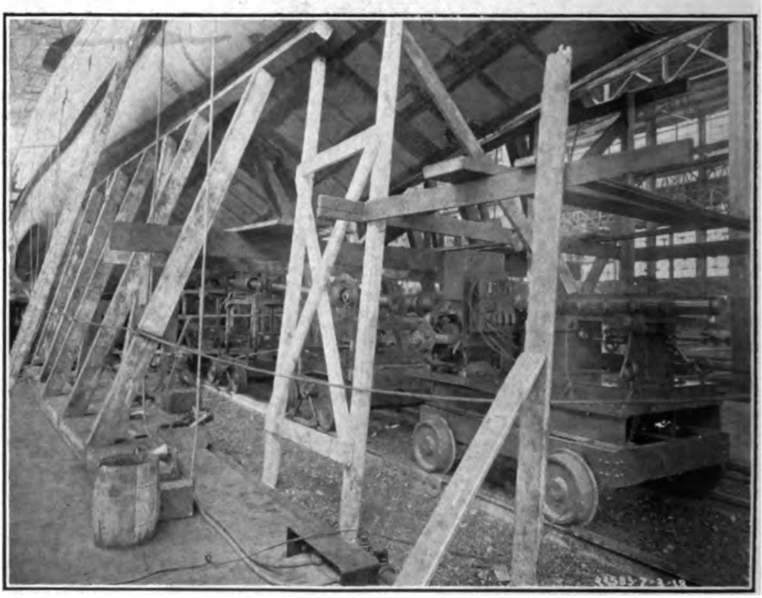


PLATE XIII.—BORING STERN AND STRUT CASTINGS.

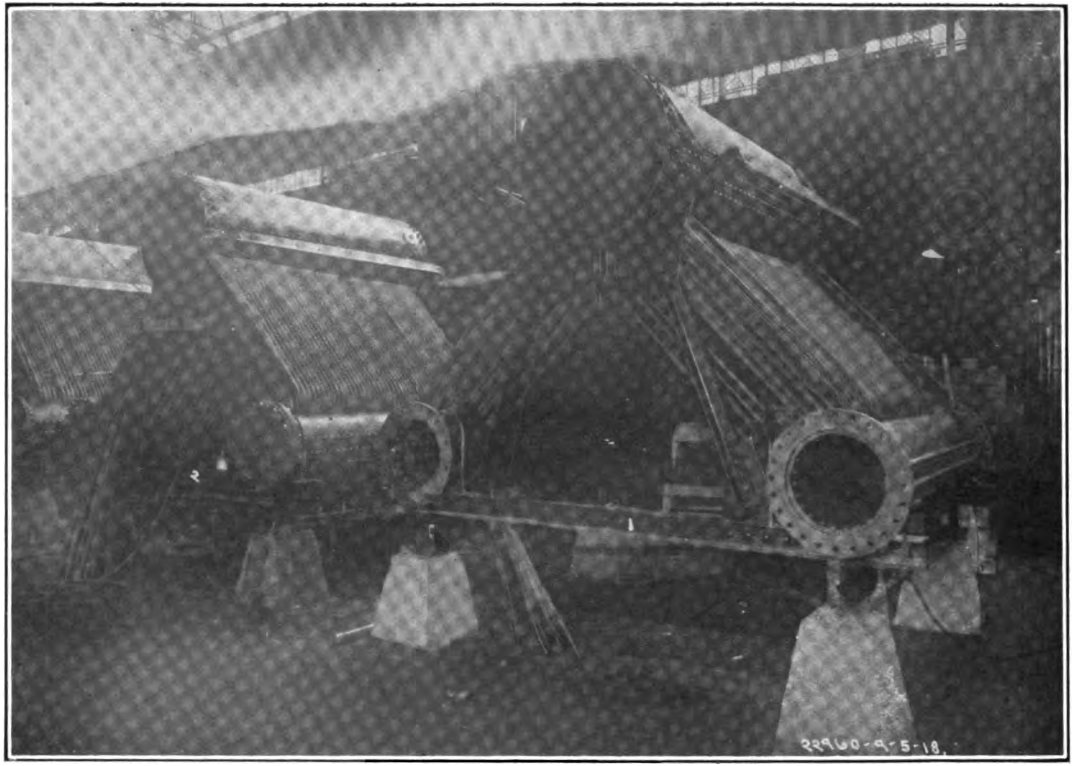


PLATE XV.—EAGLE BOAT BOILER.

and threaded, and fitted with a nut with a key-way, for securing the linear expansion coupling flange.

The forward end of the reduction gear sun pinion shaft is of the same construction and has a corresponding linear expansion coupling flange. These coupling flanges are connected by twelve steel pins which are secured in the reduction gear flange, and which work in bronze sleeves fitted in the rotor flange. A space of 3/16 inch is left between coupling flange faces, thereby allowing for linear expansion of the rotor shaft and reduction gear sun pinion shaft.

(For details of blading see Plate IX.)

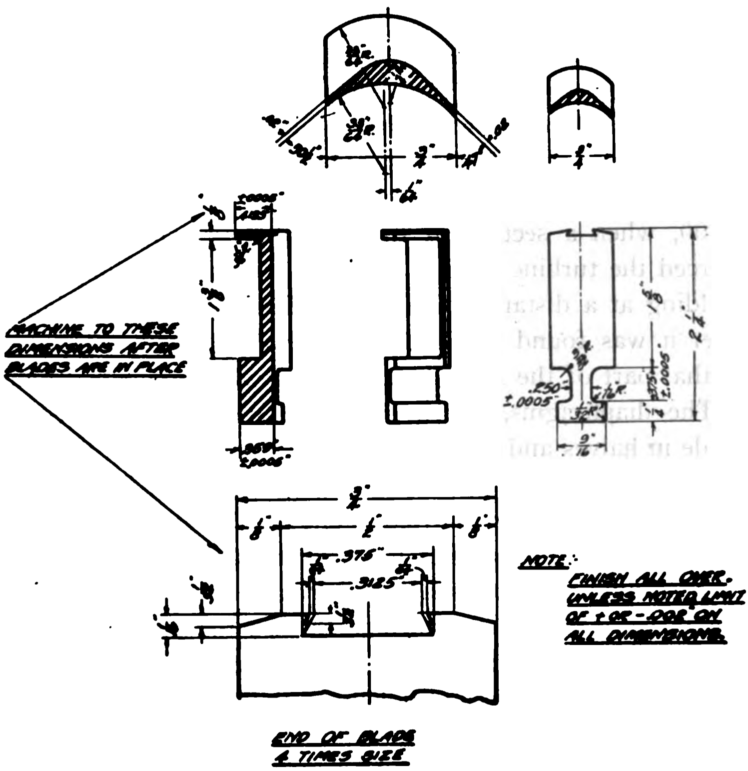


PLATE IX.—TURBINE BLADING.

The periphery of the rotor wheels are ⊥ slotted. The roots of the rotor blades are of a corresponding cross-sectional shape. A starting place for inserting the roots of the blades is provided, and, after all blades are inserted in the rotor slot, a locking piece is inserted and electrically welded.

The distance between the blades is maintained by the shapes of their roots and tips. After all the blades are assembled on a wheel, a dove-tailed slot is cut in their periphery and a flat strip of soft steel, in lengths of not over 10 inches, is then peened into this slot and electrically welded. This serves as a shroud ring. The tips of the blades are then ground to smooth them up and bring the assembled rotor wheel to its finished outside diameter.

The construction is exceptionally strong, as was evidenced through an accident in June, 1918. A block test of a turbine running at full power was being conducted, and, through error, the load was suddenly thrown off. The rotor's revolutions increased rapidly from 4,200 revolutions per minute to about 8,000, when a section of the second-stage wheel broke off, pierced the turbine casing, and then struck the ceiling of the building at a distance of about 60 feet. Upon investigation later it was found that the blading was perfectly tight, even in that part of the second-stage wheel which remained intact.

The diaphragms, which carry the stationary nozzles, are made in halves and are set in slots in the upper and the lower halves of the casing. The lower diaphragm may be rolled out when the top casing is lifted. The diaphragms are all fitted with radial gland packing to prevent leakage from stage to stage around the rotor shaft. There are six nozzles, bolted to the forward end of the turbine casing, for delivering steam to the ahead turbine. Steam is admitted to the nozzles by three balanced valves under control of the maneuvering gear, which consists of three cams on a cam rod operated by the maneuvering wheel through a system of levers and gears. Each valve admits steam to two nozzles. Steam is admitted

to the astern turbine nozzles by a 4-inch double-seated balanced valve under control of the maneuvering gear.

A double-seated balanced steam-stop valve, operated from the working platform, is fitted between the steam chest and the main steam line. A foot trip gear is installed so that this valve may be closed quickly in case of emergency. A centrifugal ring governor is installed on the forward end of the rotor shaft. This operates a trip gear which disengages a ratchet that holds the stop valve open and allows the valve to close when the revolutions of the rotor exceed a predetermined number.

Auxiliary exhaust steam may be admitted to the first pressure stage. This is controlled by a check valve and a stop valve operated from the working platform.

Projections on each side of the lower half of the casing are secured by holding down bolts to the engine foundation. The forward end of the turbine casing rests on a pedestal which allows for a sliding seat.

REDUCTION GEAR.

(See Plates X and XI.)

The reduction gear casing is of cast iron and consists of five parts in the upper half and two parts in the lower. Projections in the lower half provide feet for securing the casing to its foundation.

The reduction gear casing contains the reduction gears, spider shaft, sun pinion shaft and bearing, linear expansion shaft coupling, after turbine shaft bearing, spider shaft bearings, and a Kingsbury thrust bearing.

The elements composing the planetary reduction gear are as follows:

An internal right and left hand herring-bone or helical gear of 37 inches pitch diameter with 222 teeth. The right and left helical ring gears are set in a recess in the gear casing and keyed together and in place.

Revolving inside and in mesh with the internal ring gear are three planetary gears with the same type teeth as the ring gear. These planetary gears are 16 inches pitch diameter and have 96 teeth. They revolve on pin journals, carried on a revolving steel spider.

The planetary gears also mesh with a central sun pinion right and left hand helical gear whose shaft carries the after expansion coupling at its forward end. The forward spider bearing which is 10 inches in diameter is hollow in order to allow the sun pinion shaft to pass through.

The sun pinion shaft is connected to the after end of the main turbine shaft by the expansion coupling. The sun pinion gear, revolving the planetary gears around their pins and the internal ring gear, causes the planetary gears to give the spider with its shaft a reduction in the number of revolutions.

A Kingsbury thrust collar is fitted on the after end of the spider shaft; the thrust blocks are set in the after part of the reduction gear casing.

All bearings of the turbine and reduction gear are lubricated by the forced feed system. A pressure of about 40 pounds is carried on the after 10-inch spider shaft bearing and the 4-inch pinion shaft bearing. The high-pressure oil lubricates the 4-inch pinion shaft bearing and then passes through a hole in the shaft to the pinion gear. Oil holes are drilled in the pinion gear to communicate with the oil hole in shaft, and the lubricating oil is distributed to all the gear teeth from the pinion gear. High-pressure oil from the after 10-inch spider bearing passes through the spider shaft journal to oil passages in the spider which leads to the oil chambers in pin journals of the planetary gears.

Lubricating oil at about 6 pounds pressure is led to all other bearings.

All main turbines and reduction gears were manufactured at the Ford Motor Company's Highland Park Plant. As each turbine and gear was completed, it was assembled as a

unit and put on a test block and given a running test, at least four hours of which was at full power. The load was taken by a 1,500 kilowatt motor generator. The main engine auxiliaries—condenser, condensate pump, radojets, lubricating oil pumps and lubricating oil coolers—were used in connection with this test. Plate XII shows this installation. (For details of block test see Table I.)

TURBINE DATA.

Number of rotor wheels in ahead turbine	6
Number of rotor wheels in astern turbine	1

First Wheel.

510 blades, blade opening, inches	1,375
Width of blade, inch	.75
Length over all, inches	3.21875
Thickness of shrouding, inch	.413
Thickness of blade base, inch	.359

Second Wheel.

217 blades, blade opening, inch	.5625
Width of blade, inch	.75
Length over all, inches	1.4375
Thickness of shrouding, inch	.474
Thickness of blade base, inch	.433

Third Wheel.

217 blades, blade opening, inch	.875
Width of blade, inch	.75
Length over all, inches	1.84375
Thickness of shrouding, inch	.529
Thickness of blade base, inch	.473

Fourth Wheel.

217 blades, blade opening, inches	1.5625
Width of blade, inch	.75
Length over all, inches	2.4375
Thickness of shrouding, inch	.488
Thickness of blade base, inch	.418

TABLE I.

Power Department. FORD MOTOR COMPANY. Highland Park Plant.

Steam Turbine Test.

No. 2 turbine fitted with one-inch nozzles instead of regular ⅝-inch nozzles as no high-pressure steam was available.

Date	July 16, 1918, A.M.					
Hour	4—5	5—6	6—7	7—8	8—9	9—10
R.P.M.	4,320	4,320	4,320	4,320	4,320	4,320
E.H.P.	1,980	2,006	1,985	2,042	2,291	2,490
Steam pressure (gauge)	154.5	153.5	151.0	154.0	166.0	176.0
Steam temperature (degrees F.)	396.0	399.0	403.0	403.0	409.0	424.0
Superheat (degrees F.)	27.5	31.5	36.5	35.7	35.2	46.1
Barometer	29.32	29.32	29.32	29.30	29.30	29.30
Vacuum (inches of mercury)	27.5	27.4	27.4	27.39	27.26	27.20
Absolute pressure in condenser (inches of mercury)	1.82	1.92	1.92	1.91	2.04	2.10
Water condensed per hour (pounds)	25,500	25,800	25,500	25,800	29,400	29,880
Water condensed per e.h.p. hour	12.85	12.72	12.85	12.60	12.80	12.01
Oil used	Ursa.	Ursa.	Ursa.	Ursa.	Ursa.	Ursa.
Total oil to turbine and gears (pounds per hour)	29,300	29,550	29,200	29,900	29,200	28,410
Oil to turbine forward bearings (pounds per hour)	8,400	8,200	8,400	8,500	8,200	8,200
High pressure to gears only (pounds per hour)	13,150	13,250	13,400	13,600	13,000	12,800
Total oil less forward bearing (pounds per hour)	20,900	21,350	20,800	21,400	20,700	20,210
Temperature of oil to gears and bearings (degrees F.)	132.0	132.0	132.0	134.0	134.5	135.0
Temperature of oil from gears and bearings (degrees F.)	157.0	157.5	158.0	158.0	157.5	158.75
Temperature rise through gears and bearings	25.0	25.5	26.0	24.0	23.0	23.75
B.T.U. per hour absorbed by gears and bearings except forward bearing	260,400	271,100	270,100	257,000	234,000	218,000
H.P. loss—Gears and bearings (except forward bearing)	103.5	105.2	105.0	100.0	90.6	84.5
Efficiency of gears and bearings	95.0	95.25	95.5	95.5	96.25	97.0
Weight of oil through coolers	29,450	29,700	29,350	30,050	29,350	28,560
Temperature of oil to coolers (degrees F.)	152.0	152.0	152.5	153.0	153.8	154.0
Temperature of oil from coolers (degrees F.)	132.0	132.0	132.0	134.0	134.5	135.0
Temperature drop through coolers (degrees F.)	20.0	20.0	20.5	19.0	19.3	19.0
B.T.U. absorbed in coolers per hour	294,000	297,000	300,000	286,000	279,000	273,000
Quantity water through coolers (pounds per hour)	59,500	61,900	60,050	56,100	56,250	59,500
Temperature of cooling water	66	66	66	66	66	66
Room temperature	81	80	81	81.5	81.5	82.3

Note.—150 pounds of oil per hour to pedestal bearings added on for oil coolers.
Turbine direct connected to motor generator set.

/s/ W. F. Kendrick. 7/10/18.

Fifth Wheel.

217 blades, blade opening, inches	3.0
Width of blade, inch	.75
Length over all, inches	3.875
Thickness of shrouding, inch	.509
Thickness of blade base, inch	.397

Sixth Wheel.

243 blades, blade opening, inches	4.625
Width of blade, inch	.75
Length over all, inches	5.5
Thickness of shrouding, inch	.476
Thickness of blade base, inch	.334

In the periphery of the rotor wheel is a tee slot in which the blades are secured.

DIAPHRAGMS.

Second Stage.

56 blades—

Pitch of blades, inches	1.79
Diameter of diaphragm, inches	34.498

Third Stage.

56 blades—

Pitch of blades, inches	1.79
Diameter of diaphragm, inches	34.748

Fourth Stage.

56 blades—

Pitch of blades, inches	1.79
Diameter of diaphragm, inches	35.498

Fifth Stage.

56 blades—

Pitch of blades, inches	1.79
Diameter of diaphragm, inches	36.998

Sixth Stage.

48 blades—

Pitch of blades, inches	2.094
Diameter of diaphragm, inches	38.498

All diaphragms are bored to receive packing ring 11½ inches outside diameter.

REDUCTION GEAR DATA.

Material.

Housing enclosing gear	Cast iron.
Spider	Steel casting.
High speed pinion shaft	Chrome Vanadium steel.
Planetary gears	Chrome Vanadium steel.
Gear ring	Chrome Vanadium steel.
Low speed shaft	Chrome Vanadium steel.
Shaft couplings	Chrome Vanadium steel.
Planetary gear pin	Chrome Vanadium steel.
Coupling pins in expansion coupling	Chrome Vanadium steel.
Oil glands	Bronze.

Gears.

Internal gear, teeth	222
Planetary gears, teeth	96
Pinion gear, teeth	30
Internal gear, pitch diameter, inches	37
width, inches	6
Planetary gear, pitch diameter, inches	16
width, inches	6
Pinion gear, pitch diameter, inches	5
width, inches	6
Helix angle of teeth	23°
Ratio of reduction	8.4 to 1

All bearings in turbine and reduction gear are of the self-aligning type setting on spherical seats.

RATES OF TURNING.

Turbine shaft, revolutions per minute	4,000
Sun pinion gear, revolutions per minute	4,000
Planetary gear on pins, revolutions per minute	1,725
Line shaft and propeller shaft, revolutions per minute	475

SHAFTING LINE.

Two sections of line shaft, length, feet and inches	13–3
One section of tail shaft, length, feet and inches	28–9⅝
Total length of shafting, feet and inches	55–5⅝
Rake per foot of shafting, inch	.849
Diameter of line shafts, inches	6.25
Diameter of tail shaft, inches	6.375

There is one right hand, cast solid, manganese bronze, 3-bladed propeller, machined to pitch and polished. Diameter, 6 feet 7 inches; true pitch, 4 feet 10 inches.

SPRING BEARINGS.

There are two spring bearings supporting the line shafting. Size of bearing, 6¼ inches by 12 inches. Lined with white metal.

LIGNUM VITAE BEARINGS.

Lignum vitae bearings are fitted at frames No. 92 and No. 106.

Lignum vitae strips, 11/16 × 2 feet 3 inches.

Bearings bored out 1/32 larger than shaft for clearance.

Zincs are fitted on both ends of strut bearings and after end of stern tube.

(See Plate XIII for method of boring stern and strut castings.)

Strut and stern castings are bored by a specially constructed, motor-driven, boring machine mounted on wheels, which run on the assembly track. Witness marks for the alignment of the boring bar are secured by reference to a weighted calibrated steel wire placed in the position of the center line of the propeller and line shafting.

MAIN CONDENSERS.

There is one main condenser of the cylindrical, straight-tube type. In the lower half of its forward end and in the upper half of its after end, the tubes are made tight by being expanded into the tube sheets. The opposite ends of the tubes are made tight by means of the usual stuffing box, corset lacing packing and ferrules.

The principal dimensions are as follows:

Distance between tube sheets, feet	7
Length of tubes as fitted, feet and inches	7–2½
Thickness of tube sheets, inches	1⅛

Tubes, number	2,084
Tubes, diameter outside, inch	⅝
Tubes, thickness, B.W.G.	16
Cooling surface, square feet	2,387
Length of condenser, feet and inches	9–8
Inside diameter, feet and inches	4–10

The condenser has the following connections on the fresh water side:

One main exhaust nozzle rectangular in shape;
One condensate pump suction;
One air ejector suction;
One auxiliary exhaust steam connection;
Trap drains;
One safety valve, water sealed, inside diameter 1½ inches;
One hand hole and cover;
One vent connection from condensate pump;
Vacuum gauge connection;
Bleeder connection;
Soda cock;
Vacuum line connections for revolution counter and quick closing stop valve pilot valve.

And the following connections on water chest:

One main injection connection on forward water chest;
One overboard discharge connection on forward water chest;
One drain on each chest;
One thermometer connection on injection piping;
Air cocks.

Vacuum is established and maintained in the main condenser by means of a turbo-driven, centrifugal condensate pump and two Wheeler radojets. The condensate pump is direct connected to a 4-horsepower Terry steam turbine of the simple impulse type. This turbine has a fly ball type governor which cuts off and admits steam according to the load. In addition, there is an emergency trip governor which closes the steam admission valve in case the speed of the turbine exceeds safe limits. The condensate pump is capable of de-

livering 30,000 pounds condensate per hour against a head of 60 feet at 3 horsepower and making 2,500 revolutions per minute. The maximum capacity of this pump is 45,000 pounds condensate per hour.

The radojets remove air from the condenser by the suction effect created by a series of steam jets. They consist of two steam ejectors working in series; the upper and the lower ejectors constituting the first and second stages, respectively.

In operation live steam from a reducing valve is delivered to the upper or first series of jets. The steam expands through these jets or expansion nozzles and gains a high velocity. The jets of steam from the first stage pass across a suction space, in communication with the main condenser, where they mix and draw air from the condenser with them.

The mixture of air and steam enters a large diverging nozzle called "the diffuser," from where it is discharged at a higher pressure than at the original suction space, into a double-passage chamber surrounding the second series of jets. The steam entering the second series of jets expands radially and leaves at a high velocity and collects the mixture in the double chamber compressing it above atmospheric pressure in an annular chamber from which it is discharged past a check valve to the feed and filter tank.

When warming up the main engine and when "standing by," the temperature of the water in the feed tank is kept under the boiling temperature by a recirculating line with a thermostatic valve.

When the water in the feed tank reaches a temperature below 212 degrees F. and for which the thermostatic valve is set, the valve opens and the partial vacuum in the condenser causes the water from the feed tank to flow through the recirculating line into the main condenser where it is cooled by the main condenser circulating water. The condensate pump discharges the water back into the feed tank along with the condensed steam.

A turbo-driven double inlet centrifugal pump supplies circulating water to the main condenser. The circulating pump is arranged to take its suction from the sea through the main injection casting or, by closing the main injection valve, the bilge suction valve may be opened and injection water taken from engine room bilge.

The main circulating pump is direct connected to a type Z N 33 horsepower Terry steam turbine and is capable of discharging 3,600 gallons of water per minute against a head of 25 feet when making 1,300 revolutions per minute.

AUXILIARY CONDENSER.

There is one 20-inch diameter cylindrical shell surface condenser, containing 225 square feet of cooling surface. The condenser is mounted upon a direct acting combined air and circulating pump, having a 5-inch steam, a 6-inch air, and a 6-inch water cylinder. There is a spring-loaded valve on the auxiliary exhaust line to the condenser for regulating the back pressure.

FEED AND FILTER TANK.

The feed and filter tank is located in the forward starboard side of the engine room. It is built of steel plate 3/16 inch thick. A perforated plate separates the filter tank from the feed tank proper. The filter tank has the following connections: One condensate pump discharge, one air ejector discharge, drain pipes from coils and traps. The feed tank has the following connections: Feed suction, drain line from auxiliary machinery, drain cock, vapor escape, water gauge glass fittings, boiler compound connection to compound tank, and thermometer.

FRESH WATER SYSTEM.

The ship's tanks have a capacity of 2,400 gallons. Filling lines are installed from ship's side and from the distiller fresh-

water pump. Water is supplied to gravity tanks by a 4-inch horizontal hand pump located in the engine room.

A 75-gallon gravity tank is located in the galley. Branches run to a faucet for galley use and to a faucet outside the galley for bucket use. Officers' state rooms are provided with running water from the galley fresh-water tank. There is a 25-gallon gravity tank in the crew's work room aft.

MAIN AND AUXILIARY FEED SYSTEM.

There are two Blake-Knowles simplex, vertical, double-acting, reciprocating main feed pumps (9 inches $\times$ 6 inches $\times$ 12 inches), secured to a heavy standard in the forward starboard side of the engine room. The feed pumps take their suction from either the feed tank, the reserve feed tank, the fresh-water tank, or main condenser through a double-suction manifold. The steam lines to the pumps are fitted with Ideal pressure governors, and there is a spring-loaded relief valve on the discharge chamber of each pump.

The discharge lines from the pumps connect into a single 3-inch copper line, which leads to a No. 12 Reilly heater secured to the forward engine room bulkhead. (This heater is capable of heating 31,000 pounds of water per hour from 80 degrees F. to 215 degrees F., using exhaust steam at 10 pounds, per gauge.) The main feed line passes from the feed heater through the forward engine room bulkhead into the fire room. Here the feed line branches into two 3-inch lines, with a 3-inch high pressure gate valve on each branch. The right hand is the main feed line, and from a tee it branches into two 2-inch lines which lead to the main feed checks. From a Y on the left 3-inch line, two 2-inch lines lead to the auxiliary feed checks.

LUBRICATING OIL SYSTEM.

The forced feed lubricating oil system consists of following equipment:

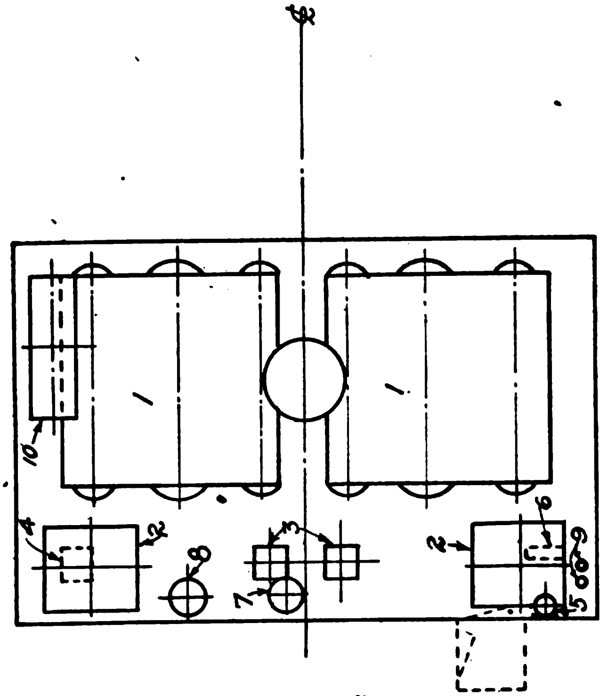
One oil storage tank of 250 gallons capacity;
Four oil return tanks of about 114 gallons each;
One oil treating tank, three sections, fitted with heating coils, filtering cartridges and a settling space;
Two Kinney turbo-driven rotary oil service pumps;
One twin oil strainer in discharge piping;
One strainer of the McComb type in the suction piping to service pumps;
Two vertical multi-whirl oil coolers;
One 4½-inch × 5-inch × 6-inch single vertical double-acting oil cooler pump;
Electric flometers on each discharge lead to bearings.

If oil stops flowing for any reason, the trouble is indicated by a light on the gauge board. Each discharge lead to bearings is fitted with a Pitot tube which shows the volume and the pressure of the oil flowing in the pipe.

BOILERS.

There are two oil-burning boilers of the Bureau of Steam Engineering express type, arranged in pairs in one compartment, as shown on Plate XIV. Plate XV shows a boiler without casing or fittings. Nearly all of these boilers were built at the Ford Motor Company's Highland Park Plant. These boilers were assembled with their casings, brickwork and fittings on the fit-out dock. They were then picked up with a locomotive crane and lowered into place.

Each boiler has 24 rows of tubes, 12 on each side. The first two rows of tubes next to the combustion chamber are 1⅜ inches outside diameter and the others are 1⅛ inches outside diameter. There are 114 of the former and 950 of the latter. There is a fore-and-aft division plate inside the smoke pipe extending from the uptake to within a few feet of the top of the pipe. This is necessary in order that work may be done in one boiler while steam is being kept up in the other.



1- BOILERS
2- FORCED DRAFT BLOWERS
3- FUEL OIL SERVICE PUMPS
4- FUEL OIL BOOSTER PUMP
5- FUEL OIL STAND-BY PUMP
6- FUEL OIL HAND PUMP
7- FUEL OIL HEATER
8- AIR COMPRESSOR
9- FOAMITE FIRE EXTINGUISHER
10- COMPRESSOR AIR TANK

U.S.S. EAGLE 1 TO 60
ARRANGEMENT OF MACHINERY PLATE XIV
IN BOILER ROOM

BOILER DATA.

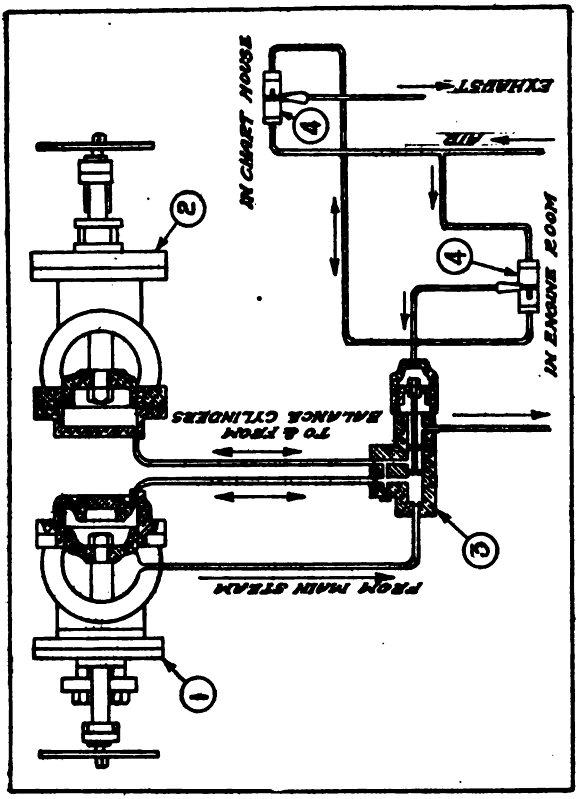
Number	2
Working pressure, pounds	250
Height to top, external, feet and inches	10–1½
Length at floor plates, feet and inches	7–8½
Width at floor plates, feet and inches	9–10
Number of furnaces to each boiler	One.
Furnace volume—each boiler, cubic feet	163
Capacity when full, water at 80 degrees, gallons	778
at steaming level, gallons	566
Heating surface—each boiler, square feet	1,500
Area through smoke pipe, square feet	9.6
Kind of draft	Closed fire room.
Atomizers—number each boiler	3
Diameter of main stop, inches	3½
auxiliary stop, inches	2½
main feed stop and check, inches	2
auxiliary stop and check, inches	2
surface blow valve, inches	1¼
bottom blow valve, inches	1½
safety valve (two) duplex—each, inches	2½
Height of funnel above floor plates, feet and inches	43–2¾
Air pressure carried at full power, inches	8
Steam space, each boiler, cubic feet, about	32
Water surface in each drum at steam level, square feet, about	25½
Type of atomization	Navy standard, pressure.

MAIN AND AUXILIARY STEAM PIPING.

Main steam piping is seamless drawn steel 203 mils thick: all joints are Van Stone. A 3½-inch main steam pipe from each boiler combines in the fire room into one 5-inch line which runs aft to the turbine. There is a bulkhead stop on this line in the engine room.

A 2¾-inch auxiliary steam pipe from each boiler combines into one 3¼-inch line in the fireroom which leads aft through the engine room bulkhead where it branches and makes a loop around the engine room. There is a stop valve on this line at the engine room bulkhead.

There is an automatic quick-closing stop valve on the main and on the auxiliary steam piping in the engine room just



U.S.S. EAGLE 1 TO 60
DIAGRAM OF QUICK CLOSING VALVE PLATE XVI.

1 – 5" MAIN STEAM VALVE
2 – 3½" AUX. STEAM VALVE
3 – PNEUMATIC VALVE
4 – THREE WAY COCK

abaft the stop valves, which, when operated, will shut steam off all machinery in the ship. (Plate XVI shows a line sketch of these valves and piping.)

The automatic quick-closing valves are of the balanced type with a piston working in a cylinder at the end opposite the valve stem. The valve stem and wheel is similar in construction to the regular navy automatic stop valve. The valve wheel is secured to a threaded bushing working in a threaded cross-piece supported on studs in the valve bonnet. The end of the valve stem works inside this bushing. When a key is inserted through the bushing and valve stem, the valve can be opened and closed in the usual manner. When the key is withdrawn the valve is under automatic control from a station in the engine room or the pilot house.

These valves are automatically opened by admitting steam pressure to the back of the balance piston. The valve is automatically closed by placing the back of the balance piston in communication with the main condenser vacuum. This is accomplished by a small master or pilot valve operated pneumatically from the control stations. The pilot valve is operated by a small air piston. When the operating valve in the engine room or pilot house is turned to a given position, air is admitted to the air piston. The air piston moves the pilot valve so that the steam pressure is cut off the back of the balance piston in the quick-closing valve. At the same time the pilot valve places the back of the quick-closing valve piston in communication with the main condenser. The partial vacuum back of the balance piston causes the valve to close. The valve remains closed as long as pneumatic pressure holds the pilot valve in this position. When it is desired to open this valve, the control valve at the operating station is placed in such a position that the air pressure behind the pilot valve air piston is released to atmosphere. A small steam piston which always has steam pressure on it now predominates and moves the pilot valve back to its original position, which cuts off

communication between the back of the balance piston and the main condenser, and, at the same time, admits steam pressure to the back of the balance piston. This steam pressure causes the valve to open.

FORCED DRAFT BLOWERS.

The forced draft installation consists of two horizontal special type Z N Terry two-bearing turbines, with extended shafts, mounting 26-inch semi-silent Green fans. The turbine is designed for 250 pounds maximum pressure and 10 pounds back pressure, and will deliver 19 horsepower at about 1,860 revolutions per minute, which is the capacity required by the Green fans for 7,500 cubic feet per minute.

AUXILIARY EXHAUST PIPING.

The auxiliary exhaust line is a 5-inch copper pipe which makes a loop around the engine room with branches to the auxiliaries. It also has connections to the main condenser, auxiliary condenser, the atmospheric exhaust line, first ahead pressure stage of the main turbine, feed water heaters, and steam coils of the evaporators. There are spring-loaded valves in this piping to the main condenser and to the auxiliary condenser for regulating the back pressure. All sections of this piping have Van Stone joints.

FUEL-OIL SYSTEM.

This system consists of one booster pump, two fuel-oil service pumps, one auxiliary fuel-oil service pump, a Reilly fuel-oil heater, valves and piping as required. The service pumps are the Kinney, turbo-driven, rotary plunger type. The fuel-oil suction piping is so arranged that the fuel-oil can be pumped from any one compartment to any other, or overboard, with the booster pump. The service pumps take their suction from the booster pump or direct from the fuel oil compartment. Suction piping has Van Stone joints. All pressure piping has flanged metal to metal joints.

The auxiliary fuel-oil service pump is a 4-inch × 2½-inch × 5-inch simplex, direct, double-acting, reciprocating pump, manufactured by the Union Steam Pump Company, installed on starboard side aft in fire room for port use. It has a connection to fuel oil suction manifold and discharges into the fuel oil discharge line.

AIR COMPRESSOR.

There is one vertical, steam-driven, direct-acting 11-inch × 11-inch × 12-inch Westinghouse standard air compressor installed on the port side of the fire room, with a 24⅜-inch × 78-inch air reservoir. Pneumatic lines lead from the reservoir through the machinery spaces with convenient connections for blowing boiler tubes, air atomization for galley range burners, piping to quick-closing stop valves and for operating pneumatic tools.

EVAPORATING AND DISTILLING PLANT.

This consists of:

Two high-pressure, single-effect Reilly evaporators of the vertical, submerged, multicoil type;

One vapor separator;

Two vertical, multicoil distillers;

One distiller test tank of 50 gallons capacity;

One 3½-inch × 4-inch × 4-inch distiller fresh-water pump;

One 7-inch × 7-inch × 12-inch distiller circulating pump;

One 3½-inch × 4-inch × 4-inch evaporating feed pump;

One evaporator feed water heater.

The rated capacity of the plant is 2,500 gallons per day. The system is provided with reducing valves on steam to coils, and traps to take care of coil condensate. Provision is made for supplying steam coils with steam from auxiliary exhaust piping when working at reduced capacity.

DATA.

One Evaporator.

Type	No. 6 Reilly, vertical, multicoil (submerged).
Diameter, inside, inches	22
Height over all, feet and inches	5–11⅛
Number of coils	6
Diameter of coils, inch	1
Thickness of coils	No. 16 B.W.G.
Heating surface, square feet	16.1
Diameter of steam nozzle, inches	1½
vapor nozzle, inches	2½
feed valve, inch	¾
blow valve, inches	3

One Distiller.

Type	"D" No. 5 Reilly, vertical, multicoil.
Diameter, inside, inches	16½
Height over all, feet and inches	4–3
Number of coils	5
Diameter of coils, inch	1
Thickness of coils	No. 16 B.W.G.
Cooling surface, square feet	19.85
Diameter of vapor inlet, inches	2½
Drain connection, inch	1
Circulating water connection, inches	2

Evaporator Feed Water Heater.

There is one type "D" No. 2 Reilly, vertical, multicoil feed water heater located on the engine room bulkhead above the evaporators.

Diameter, inside, inches	10
Number of coils	2
Height over all, feet and inches	4–3
Diameter of coils, inch	1
Thickness of coils	No. 16 B.W.G.
Heating surface, square feet	7.94
Diameter vapor inlet, inches	2
outlet, inches	2
Diameter feed inlet, inches	1½
outlet, inches	1½

REFRIGERATING INSTALLATION.

One Audiffren-Singrum, one-half ton. sulphur dioxide refrigerating machine with cold storage box is installed on *Eagles* Nos. 1 to 50, inclusive, in a compartment on the port side just forward of frame 50. The York, one ton C O^2, refrigerating machine is installed on *Eagles* Nos. 51 to 60, inclusive.

PNEUMERCATOR.

A pneumercator system to indicate the level of oil in the fuel oil compartments is installed in *Eagles* Nos. 8 to 56, inclusive.

STEERING GEAR.

A steering engine of the Williamson type is installed on the after engine room bulkhead. This steering engine is of the two-cylinder, differential valve type, operating at 125 pounds pressure.

The steering gear is of the right and left screw gear type. The main steering gear control is located in the chart house. A hand-steering wheel is located on the after deck house.

The steam steering gear is capable of putting the rudder from the hard-over to hard-over in twenty seconds when the vessel is steaming at full speed.

ANCHOR ENGINE.

The anchor engine is a two-cylinder, differential valve type, designed to operate at 125 pounds pressure. It is located on the main deck forward of the forward deck house.

DRAINAGE SYSTEM.

For draining the boiler room, engine room and storerooms immediately forward of the boiler room, a 3-inch galvanized pipe is led through these compartments and connected with the fire and bilge pumps in the engine room. Branch lines are taken from the drain piping and led to the lowest point of the compartments and fitted with strainers and stop-

check valves. Suctions are located in forward and after ends of the engine room and fire room. The compartment forward of the fire room has one suction. There is a six-inch suction and piping in the engine room to the circulating pump, which can be used in freeing the engine room of large quantities of water. There is a 2-inch steam ejector in the fire room with a suction in the bilge, and one in the crew's quarters aft with a suction on the platform deck. Other compartments are fitted with stand pipes with connections on the platform deck for the standard Navy handy-billy pump.

FIRE MAIN.

The fire main consists of 2½-inch galvanized piping which runs directly under the main deck from frame No. 31 to No. 95. It connects to two 7-inch × 7-inch × 12-inch vertical, double-acting, simplex Worthington Blake-Knowles fire and bilge pumps located in the engine room. Cut-out valves divide the system into a forward and an after section. Risers with connections lead from the fire main so that every point in the vessel may be reached by a 50-foot length of hose. Water for the flushing system is taken off the fire main.

GALLEY.

The galley installation consists of one two-section oil-burning range, fitted to use either air or steam for atomization, two steam aluminum kettles of twenty gallons each, one coffee urn of twenty gallons capacity, a dresser with sink and steam-heating pipe, and a fuel oil tank with a hand-fuel oil supply pump taking a suction from the fuel oil manifold in the forward part of the fire room.

WEIGHTS.

Group No.	*Description.*	*Weights.*
1	Main turbine, reduction gear, Kingsbury thrust and exhaust connection	21,394
2	Shafting, including casings, sleeves and coupling bolts	5,954
3	Bearings, including bulkheads, stuffing boxes and fairwaters	1,294

6	Main condenser	16,270
7	Main circulating pump with turbine	2,050
	Condensate pump with motor and air ejectors	1,021
8	Propeller	1,550
9–10	Boilers and boiler fittings complete, including burners	50,488
11	Smoke pipes	2,655
	Uptakes	4,490
12	Steam and exhaust pipes and valves	12,504
	Condensate piping	7,329
13	Suction, discharge pipes and valves	7,848
14	Asbestos pipe covering and clothing	5,750
	Sheet metal jacketing	1,500
15	Flooring and grating	5,275
16	Auxiliaries—Main feed pumps	1,900
	Fire and bilge pumps	1,719
	Blowers	2,388
17	Fittings and gear—Feed and filter tank	1,501
	Waste, oil and other small tanks	2,512
	Lifting gear	250
	Gear for operating valves	130
	Gauges, telegraphs, etc	560
	Feed water heater	1,596
	Miscellaneous gear	250
18	Water feed tank	2,417
	condenser	3,005
	feed heater	74
	boilers	8,549
	pumps and pipes	2,175
19	Tools, stores and spare parts (not installed)	
20	Miscellaneous machinery—Air compressor and tanks and piping	1,828
	Distilling plant—evaporator	1,842
	Distillers	2,300
	Heater	625
	Pumps	1,232
	Tanks, piping, water, etc	4,334
	Heating	1,829
	Refrigerating plant, including ice-box	2,250
	Fuel oil pumps, piping, heaters, etc	7,209
	Lubricating oil pumps, piping, coolers, etc	4,418
	Auxiliary condenser and pump	2,952
	Standby pump	188¼
	Fuel oil suction piping to aft tank	273
	Piping, lagging and fitting on auxiliary condenser lines	1,229
	Total, pounds	**208,907¼**

ELECTRICAL INSTALLATION.

There are two General Electric 10 kilowatt turbo generators. direct connected, 125 volts, 4,000 revolutions per minute. The turbines are designed to operate on full boiler pressure and 10 pounds back pressure.

The main switchboard, distribution panel, and battery charging panel are combined and are located near the generating sets in the engine room. There are three lighting circuits: general, battle and auxiliary. The wiring is Navy Standard.

The auxiliary lighting circuit is arranged to cut in automatically when the quick-closing stop valves are thrown. This circuit is energized by two sets of storage batteries, 32 volts, 210 ampère hour capacity. The auxiliary lights are distributed in parts of the ship where light is essential in case of emergency.

There are two ⅛-horsepower, 32-volt, ventilating sets for forced ventilation. They are operated normally on the 125-volt circuit through a series resistance, which is automatically cut out by relays when it is desired to run on auxiliary circuit.

The radio installation consists of a 1 kilowatt Navy Standard telegraph set and a radio telephone set. Both sets can be controlled from the pilot house. The radio telegraph motor generator is located in the engine room. The telephone is energized from the battery circuit.

In addition to the counter gear, there is an electric tachometer, mounted alongside the gauge board, which indicates the number of revolutions the propeller shaft is actually making and the direction of rotation. The tachometer magneto is driven from a split gear on the propeller shaft.

Reports which have been received concerning the operations of *Eagles 1, 2* and *3* which have been doing duty in the White Sea state that those boats have been doing despatch duty between Murmansk and Archangel, each cruising about 8,000

miles at a standard speed of 14 knots in waters full of floating ice, that they have satisfactorily carried out target practice at a speed of 17 knots, and that they are satisfactory in all respects.

REMARKS ON THE PROBLEM OF IMPACT.

BY N. W. AKIMOFF, ASSOCIATE.

The problem of impact contains many elusive points. In an average text book it covers but a few pages of space, and we are started off with the following hazy notions:

1. We know something about the coefficient of restitution.
2. Carnot's theorem is given to us as a firm basis for solution of almost any problem: *kinetic energy, lost in impact, is equal to that, due to lost velocity.* This, as a rule, is not so at all, in ordinary problems of life, although, in itself, from the standpoint of theory, Carnot's theorem is, of course, quite correct.
3. We also retain, too well, as a rule, that in impact something is just twice the value it would have if the force were applied gradually. Whether this *something* is the stress or the deflection, and what the other conditions of the problem are, does not matter; but the ratio itself, *twice,* survives longer, generally, than any other notion acquired in the school; and we are actually apt to hypnotize ourselves into the belief that a weight of 2 pounds if dropped on the floor from the height, say, of 5 feet, will produce the *force of blow* exactly equivalent to a 4-pound weight gently placed on the said floor. Anyone who wishes to be cured from this absurd notion is invited to experiment by placing, and then dropping, the weight on his toes; he will see the point at once.

In general, the idea of *force of blow,* and the rather closely associated one, of *work of blow,* mean *absolutely nothing;* in themselves, such things do not exist. It is like that other commonplace question one is often asked, in reference to a wound music-box spring: *how many horsepower is there in such a spring?* There can positively be no answer to such a

question in itself, unless one knows the rate at which the work will be given out by the spring.

The object of these few pages is to illustrate the above in an elementary manner. Some of the arguments have been

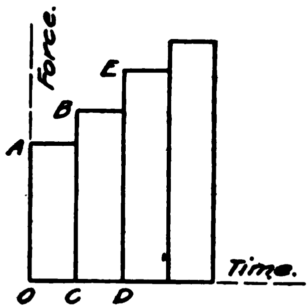


Fig.-1.
Momentum as area.

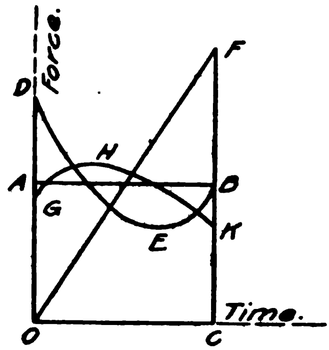


Fig.-2.
Various manners in which the same momentum can be produced.

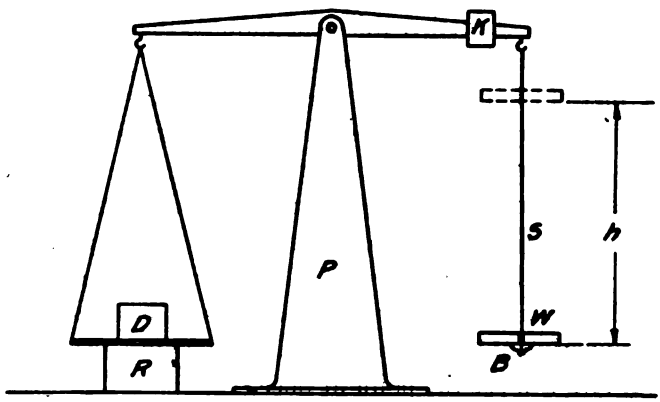


Fig.-3.
Prof. Bouasse's experiment.

adapted from Bouasse's "Théories de la Mécanique" (Paris, 1895), while the drop experiments have been performed in writer's Laboratory, by his assistant engineers, Messrs. F. G. Hechler and G. W. Klinger.

It is a well-known theorem of dynamics, that $Ft = mv$, in other words, the product of the force by the time, during which the force acts, is equal to the momentum acquired by the body (or to the increase in momentum, if the body was in motion to begin with).

The graphical illustration of this principle would be as follows: If forces are laid off as ordinates (Fig. 1) and time on the axis of abscissae, it is evident that the area of the rectangle A-C will represent to a certain scale the momentum imparted to the body by the force OA which has acted through the time-interval OC. Similarly, the area of the rectangle BD will represent an increase of momentum, due to the (suddenly increased) force, BC, acting during the time-interval CD, etc. This can, of course, be extended to gradually varying forces, in which case the upper boundary would be a smooth curve. In other words, momentum can always be represented as an area; hence the conclusion, that, being given a certain amount of momentum (a certain area) we are at liberty to imagine a great variety of combinations of force with time, each of which would answer.

For instance, if the momentum, acquired by the body (or lost, for that matter) is expressed as the rectangle OB (Fig. 2), this may have been due:

1. To the constant force OA, acting during OC seconds.
2. To the decreasing force, varying along DEB.
3. To the uniformly increasing force OF.
4. To the variable force GHK,

etc., with the only proviso, that the area OB is maintained.

Furthermore, we do not have to limit ourselves to the same time element OC, but can readily imagine that the same area, OB, can be built up on a very short time interval, which, of course, would bring into play much greater forces. Thus, in order to acquire the velocity of 2,500 feet per second, a shell must be let fall during practically 78 seconds; the same shell, acquiring the muzzle velocity of 2,500 feet per second,

in say, one one-hundredth of a second, must be acted upon by a force 7,760 times the weight of the shell (assuming, for the sake of illustration, the latter force to be constant, which means that the *maximum* force may have been even *greater*). It is quite clear, therefore, that the action of an instantaneous force (which, in reality, mean quick-acting force, and not mathematically instantaneous one) in no manner differs from that of a force acting slowly; only the magnitude of the latter will naturally be high, because the momentum area is to be maintained.

Now, as to the force of blow: the great French philosopher Descartes wrote, as far back as 1640, the following lines in a letter to a friend: "To determine how much weight would be required to equal the force of a hammer blow, is a question of fact, where pure reasoning is of no avail without experiment."

Such an experiment, in the form suggested by Prof. Bouasse, can be performed as follows (Fig. 3): The right scale-pan of the scale P has been removed, and its weight made up by means of the counterweight K; on the string S is suspended a button B and a sliding weight W. The left pan is loaded with the weight D which is heavier than the combined weight of W, S and B, so that the left pan is in contact with the rest R. If there is such a thing as *force of blow*, it is natural to suppose that the weight W, lifted through a certain height h, and then dropped on the button B, thus producing the effect of a force, will lift, momentarily, the left pan from the rest. As a matter of fact, we observe nothing of the sort: the height h, alone, does not mean anything definite; the effect, the *force of the blow* depends on the elastic properties of the string S. If the height h has been found for an inextensible string, it will not produce the desired effect if a yielding string is substituted. In other words, the *momentum itself* tells us absolutely nothing with regard to the force of the blow, unless we prescribe other conditions as well.

In some of the text books the conditions prescribed are such that they *positively cannot* be met with in practice, and a great deal of harm is thereby accomplished to the student's mind. With regard to the action of impact on beams, the following general remarks may be made: if the load W is gradually applied, the deflection will increase gradually, and will reach some final value f; if *the same load* is applied suddenly, the deflection will momentarily reach the value 2 f, and, of course, the corresponding stress will be twice the value due to gradual application; however, the beam will quiet down, after a few vibrations, to the same value f as before, and the stress will be decreased accordingly, *i.e.*, to one-half of the maximum value. All this applies to the sudden application of the weight W, *not of any greater load*, and *not* to dropping of W *from* any height; for instance, let the load W rest in contact with the beam, while its actual weight is taken up by a string; if now the string is cut, we have the case of a sudden application of the load W. To this we might properly refer as dropping from zero-height. If dropped from greater height than zero, the weight will develop much greater stress and cause a greater deflection than the values we just had. How much greater, depends on conditions (see excellent papers, by H. D. Tieman, "Journal Franklin Institute," 1909, also by Armin Elmendorf, *ibid.*, 1916).

As a rule, the following formula, given by St. Venant, gives satisfactory results:

$$f_r = f_s + \sqrt{f_s^2 + f_s \cdot \frac{v^2}{g} \cdot \frac{m}{m+n}},$$

where f_r is the maximum deflection, due to dropping of the weight.

f_s static deflection, corresponding to gradual application of the same load.

v velocity of the load when striking the beam.

m ratio of weight of the load to that of the beam.

n correction coefficient, depending on the method of fastening of the ends of the beam; also on the location of the point at which the beam is struck (St. Venant, "Élasticité des Corps Solides," p. 490, etc.).

This formula, although only approximate, means that both the inertia of the beam and the conditions of the ends have something to do with the value of the greatest deflection (and, therefore, of the greatest stress) due to the impact.

As regards plates, such as floor slabs, armor plates, etc., the matter has not been sufficiently worked out, so far; it is not unreaonable to suppose that the same methods would apply; first, we would assume that the *form* of the deflected surface would not greatly differ from that due to static load; we know enough about deflections of plates to work this out; next, we would determine the potential energy in function of this deflection; finally, the latter would be made equal to the potential energy, due to the following load. The correction, taking into consideration the end conditions and the location of the point of impact, would have to be made along the same lines as was done by St. Venant with regard to beams. All this is difficult, but it most certainly can be done.

The experiments made in writer's Laboratory comprise a set of drop tests, in which the test piece *p* (see plate) was struck by the sharp point *c*, forming part of the tup *t*; the latter was dropped from various heights, as given in the appended tables. The test piece was resting on the stand *a*, to which we shall refer as anvil, and was made of the following material: hard rubber (valve disk), lead and hard babbitt. In some of the tests the anvil was firmly secured to a heavy cast iron foundation plate; in others, the anvil was placed upon various shock-absorbing substances, such as soft rubber; insulating cork; three soft springs; the characteristics are given below. The "sharp point" C was the apex of a cone, of the following angles: 30 degrees, 45 degrees and 60 degrees. The

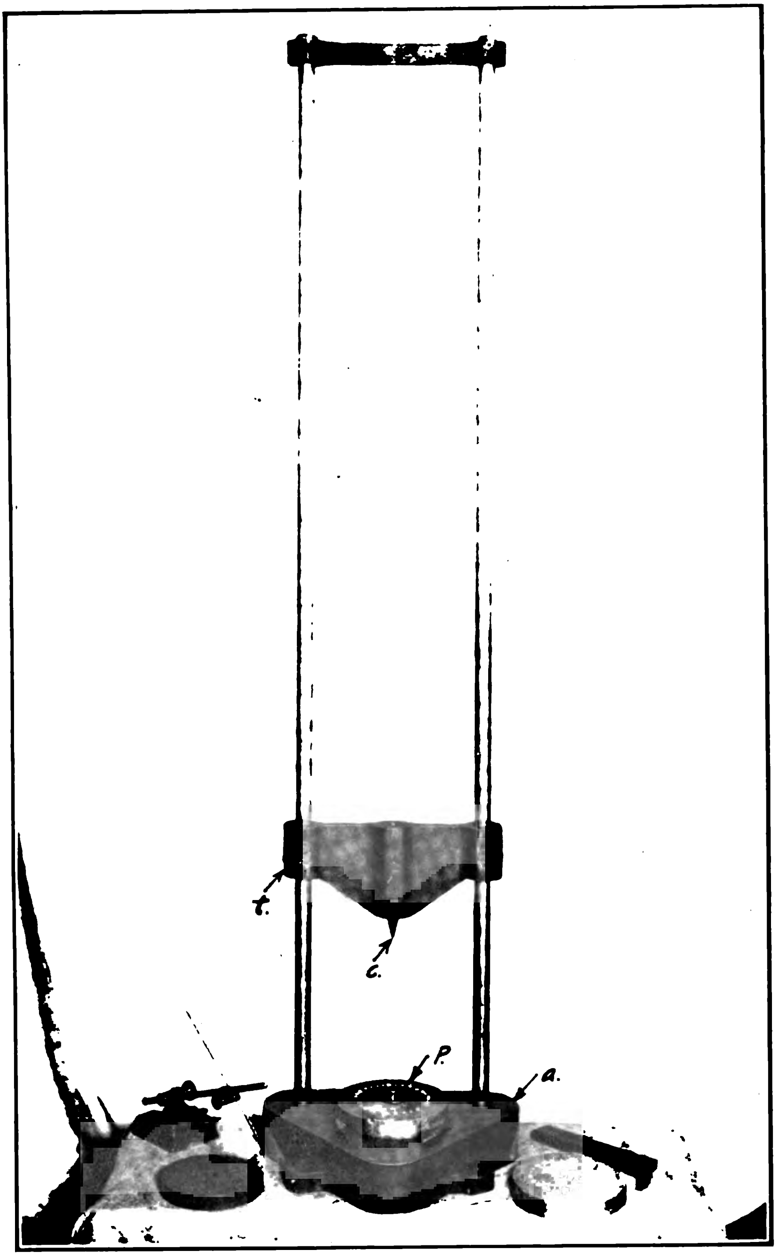


PLATE I.

Note: The figures given in this table represent "Depth of Holes" in inches.		HEIGHT OF DROP (INCHES)								
		12			24			36		
		Test Pieces			Test Pieces			Test Pieces		
"Sharp Point" Apex of Cone	Anvil mounted on various materials	Lead	Babbitt	Hard Rubber	Lead	Babbit	Hard Rubber	Lead	Babbitt	Hard Rubber
30"	Cast iron	.41	.28	.25	.53	.34	.33	.62	.44	.39
	Cork	.34	.23	.19	.48	.33	.25	.59	.45	.36
	Rubber	.29	.19	.17	.40	.27	.23	.53	.37	.31
	Springs	.28	.15	.13	.37	.26	.19	.50	.35	.23
45"	Cast Iron	.30	.22	.18	.41	.32	.27	.49	.34	.33
	Cork	.28	.19	.16	.40	.25	.23	.48	.31	.28
	Rubber	.25	.16	.13	.36	.24	.20	.44	.30	.25
	Springs	.24	.12	.08	.31	.23	.18	.41	.28	.22
60"	Cast Iron	.27	.18	.16	.36	.23	.20	.43	.27	.25
	Cork	.25	.16	.14	.33	.19	.18	.41	.25	.23
	Rubber	.22	.14	.10	.28	.18	.17	.38	.24	.20
	Springs	.19	.09	.06	.25	.17	.14	.34	.23	.19

ratio of the moving weight to that of the anvil (including the test piece) was maintained constant.

These tests are meant to be a further illustration of the same fact, *i.e.*, that, in itself, there is no such thing as *force of blow* (or *work of blow*); the apparatus shown in the plate, home-made as it is, proved to be satisfactory for this purpose. Of course, it was not designed with the object of competing with Mr. Amsler's well-known drop machine, so that the data, given in the appended table, are of qualitative interest only.

TORSIONAL VIBRATIONS OF IRREGULAR SHAFTS.

By Frank M. Lewis.

ASSISTANT PROFESSOR OF MATHEMATICS, WEBB ACADEMY, NEW YORK.

Engineers have for some years recognized that, under favorable conditions serious torsional vibrations may occur in the shafting of ships. The most important facts that have been ascertained are briefly as follows:

The turning moment of the engine is subject to periodic variations. This period is some multiple of the revolutions, depending upon the type of engine and number of cylinders. In case of doubt as to the period of the principal variations, the turning moment diagram may be analyzed by means of Fourier's Series. * * * See "Synchronous Torsional Vibrations of Shafting," by Hermann Frahm, Journal of the American Society of Naval Engineers, Vol. XIV.

Every shaft has one or more natural periods of torsional vibration. This period is for the case when the shaft is free at both ends.

When the natural period of vibration of the shaft approaches the period of variation of the turning moment of the engine, the vibrations of the shaft tend to increase in magnitude, the stresses in the shaft due to these vibrations increasing until fracture may ultimately occur. Vibrations may also be caused by irregularity of the turning moment of the propeller.

In the design of shafts we should therefore avoid this synchronism and be certain before the engine is built and trouble develops that these vibrations will not occur.

To do this it is necessary to have a method for readily calculating the natural period of torsional vibration of any shaft.

The methods of calculation hitherto proposed are applicable only to a very simple case, namely, a long length of shaft to which is fastened at either end a heavy rotating mass, propeller, engine, turbine, or something similar. Turning to Fig. 4, the shafting of U. S. submarines *S-4* to *S-13*, it is seen that we here have a shaft that can not be considered as even approximating to this simple case. These methods will moreover determine only the period of first order vibrations, or those having one node. As will be seen later, those of secondary order, or two nodes, may be just as important.

We demonstrate in this article a simple graphical method for determining the natural period of torsional vibration of any shaft, no matter in how complicated a manner the distribution of the rotating masses or the stiffness of the shaft itself may vary from end to end. It will moreover determine with equal ease the period of the vibrations of secondary or even higher order, if that is required.

We will discuss first the three general properties of a shaft that determine its period. They are:

The inertia of the rotating masses.
The stiffness.
The length.

The inertia of the rotating masses is measured by the mass polar moment of inertia. The amount of this mass polar moment of inertia per unit of length will be denoted by J_w.

The stiffness of the shaft is measured by the product GJ_s where J_s is the moment of inertia of that portion of the shaft that takes torsion and G is the coefficient of elasticity for shearing. For a section of simple shaft to which nothing is attached it follows that $J_w = wJ_s$, w being the unit weight of the shaft material. If on the shaft there is fastened an armature or other mass so that J_w does not equal wJ_s, that shaft is said to be a loaded shaft.

The use of the product GJ_s as a measure of stiffness follows directly from the formula for the deflection of shafts.

$$\theta = \frac{M\,l}{G\,J_s} \qquad (1)$$

M being twisting moment.
l length of shaft.
θ torsional deflection in radians.

That is, torsional deflection under a given twisting moment and length is inversely proportional to GJ_s.

It may happen, as in way of the cranks of an engine, that GJ_s has no direct meaning. If we can determine the torsional deflection of such a shaft under any given twisting moment we can determine by formula (1) an equivalent GJ_s.

We will define a step variable as being a function which remains constant for a certain range of the independent variable and then jumps to a new constant value for the next consecutive range, and so on for its entire length. The graph of some such function might be as in Figure I.

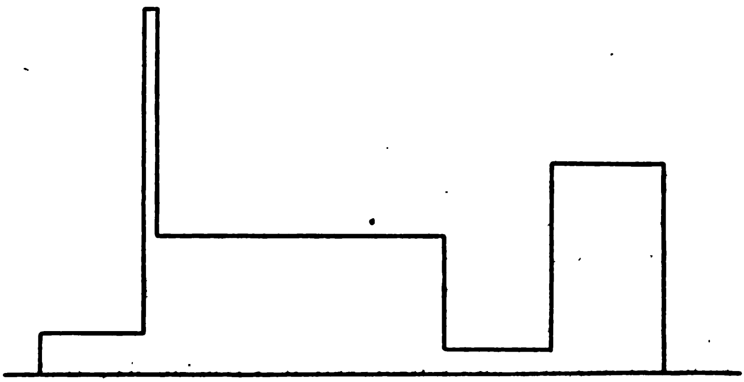


FIG. I. GRAPH OF STEP VARIABLE

By choosing suitable breaking points every shaft may be considered as being a step variable.

Over each section or step both J_w and GJ_s are considered as being constant. Unless both the shaft and its load are of uniform dimensions throughout the step, they will, of course

not be constant. In this case equivalent constant values can be calculated.

The equivalent J_w is equal to the total mass moment of inertia of the step divided by the length of the step.

The equivalent GJ_s for the step is determined by finding the total torsional deflection under a given twisting moment. Then from formula (1),

$$G\,J_s = \frac{M\,l}{\theta}$$

will be the equivalent value. The reader will note the step-page used for the shaft in Fig. 4.

The calculator must use his judgment in choosing the breaking points, so that the equivalent shaft thus obtained will closely approximate to the actual one.

In general the use of steps of too short a length will involve a great deal more labor without any compensating gain in accuracy.

We will now proceed to the mathematical demonstration of the method.

Let M = Twisting moment in shaft.
$G\,J_s$ as previously explained.
θ = Twist of shaft in radians.
x = Length variable for shaft.
g = Gravity constant.
J_w = Mass polar moment of inertia per unit of length.
l = length of any step.

Consider any step and take a section of length δx. Let the increment of twist in the length δx be $\delta\theta$. Then rewriting equation (1) we get

$$M = G\,J_s \frac{\delta\,\theta}{\delta x} \tag{1a}$$

Differentiating this we get

$$\delta M = G\,J_s \frac{\delta^2\,\theta}{\delta x^2}\,\delta x \tag{2}$$

Let the acceleration of this small section be $\frac{\delta^2 \theta}{\delta t^2}$

Then the increment of twisting moment in the length δx due to the inertia forces will be

$$\delta M = \frac{J_w}{g} \cdot \frac{\delta^2 \theta}{\delta t^2} \delta x \tag{3}$$

Equating (2) and (3) we get

$$G J_s \frac{\delta \theta}{\delta x^2} = \frac{J_w}{g} \cdot \frac{\delta^2 \theta}{\delta t^2} \tag{4}$$

or

$$\frac{\delta^2 \theta}{\delta x^2} = \frac{J_w}{g G J_s} \cdot \frac{\delta^2 \theta}{\delta t^2} \tag{5}$$

Equation (5) is in the form of the general equation for all harmonic motion. We are only interested however in the simple harmonic motions that can take place.

For simple harmonic motion the general equation is

$$\frac{\delta^2 \theta}{\delta t^2} = - k^2 \theta \tag{6}$$

If t_1 is the time of a double oscillation, then the familiar solution of (6) is given by

$$t_1 = 2 \frac{\pi}{k}$$

Or if n is the number of complete vibrations per second

$$n = \frac{k}{2 \pi} \tag{7}$$

Squaring

$$k^2 = 4 \pi^2 n^2$$

So that

$$\frac{\delta^2 \theta}{\delta t^2} = - 4 \pi^2 n^2 \theta \tag{8}$$

Substituting (8) in (5) we get

$$\frac{\delta^2 \theta}{\delta x^2} = - \frac{4 \pi^2 J_w}{g G J_s} n^2 \theta \tag{9}$$

The general solution of (9) is given by the equation

$$\theta = a \cos (\beta x + \gamma) \tag{10}$$

From which by differentiation

$$\frac{\delta \theta}{\delta x} = -\alpha \beta \sin (\beta x + \gamma) \tag{11}$$

$$\frac{\delta^2 \theta}{\delta x^2} = -\alpha \beta^2 \cos (\beta x + \gamma) \tag{12}$$

α, β and γ are constants. They must be determined for each step so as to satisfy, first the differential equation for that step, and second, the end conditions for that step. It will be noted that n, the vibrations per second, which is the answer we are seeking, enters into equation (9). It will therefore enter into the constants β and γ in every step. The value of n, which enables us to satisfy every differential equation and every end condition throughout the entire length of shafting, will be the solution of the problem which we are seeking. To simplify the explanation we will assume for the present that this value of n is known.

Substituting (10) and (12) in (9) and simplifying we get

$$\beta^2 = \frac{4 \pi^2 J_w}{g G J_s} n^2 \tag{14}$$

Or

$$\beta = \frac{2 \pi n \sqrt{J_w}}{\sqrt{g G J_s}} \tag{15}$$

We will investigate first the end conditions at the beginning of the shaft. Let there be fastened to this end of the shaft a rotating mass of polar moment of inertia J_1.

Then if M is the twisting moment at this end

$$M = \frac{J_1}{g} \cdot \frac{\delta^2 \theta}{\delta t^2} \tag{17}$$

Also

$$M = G J_s. \frac{\delta \theta}{\delta x} \tag{1a}$$

Equating

$$\frac{J_1}{g} \cdot \frac{\delta^2 \theta}{\delta t^2} = G J_s. \frac{\delta \theta}{\delta x} \tag{18}$$

From (8)

$$\frac{\delta^2 \theta}{\delta t^2} = -4 \pi^2 n^2 \theta$$

From (10) $\theta = a \cos(\beta x + \gamma)$

From (11) $\frac{\delta \theta}{\delta x} = - a \beta \sin(\beta x + \gamma)$

Substituting these values in (18) we get

$$\cos(\beta x + \gamma) \frac{4 \pi^2 n^2 J_1}{g G J_s} = \beta \sin(\beta x + \gamma) \qquad (19)$$

Since we are considering the beginning of the shaft, $x = 0$ and (19) reduces to

$$\tan \gamma = \frac{2 \pi n J_1}{\sqrt{g G J_s J_w}} \qquad (20)$$

This is the condition to be satisfied at the beginning of the shaft.

If nothing is attached to the shaft at this point tan γ reduces to zero.

We have now determined the constants β and γ for the first step in terms of n and the known properties of the shaft. a can have any value for this step. It is a measure of the amplitude of vibration at the beginning of the shaft.

We will now investigate the end conditions for the second step. We will use the subscript 1 to denote the first step, 2 the second step, etc.

From (15) $\beta_2 = \frac{2 \pi n \sqrt{J_{w2}}}{\sqrt{g G_2 J_{s2}}}$ (21)

At the end of the first step

$$M_1 = G_1 J_{s1} \frac{\delta \theta}{\delta x} \qquad (22)$$

Substituting the values for $\frac{\delta \theta}{\delta x}$ and β_2 given by (11) and (21) in (22) we get

$$M_1 = - \frac{a_1 2 \pi n \sqrt{G_1 J_{s1} J_{w1}}}{\sqrt{g}} \sin(\beta_1 l_1 + \gamma_1) \qquad (23)$$

Likewise at the beginning of the second step, since for this point $x = o$

$$M_2 = -\frac{a_2\, 2\pi n \sqrt{G_2 J_{s2} J_{w2}}}{\sqrt{g}} \sin \gamma_2 \qquad (24)$$

l_1 being the length of the first step.

The turning moments on each side of the break are equal. Therefore equating M_1 and M_2 and clearing we get

$$a_1 \sqrt{G_1 J_{s1} J_{w1}} \sin(\beta_1 l_1 + \gamma_1) = a_2 \sqrt{G_2 J_{s2} J_{w2}} \sin \gamma_2 \qquad (25)$$

Also at the break the deflections must be equal, or

$$a_1 \cos(\beta_1 l_1 + \gamma_1) = a_2 \cos \gamma_2 \qquad (26)$$

We will now show how these equations and conditions can be expressed graphically. Turning to Fig. 2 we lay off a distance OA of length equal to unity. Lay off a perpendicular AC equal to $\tan \gamma$ as given by equation (20). Then $\angle$ COA $= \gamma$ and OC $= a$.

Substituting the value of β given by (15) in (10) we get

$$\theta = a_1 \cos\left\{\frac{2\pi n \sqrt{J_{w1}}}{\sqrt{g\, G_1\, J_{s1}}}\, x + \gamma_1\right\} \qquad (27)$$

The angles are in radians. It will be convenient to put them in degrees. In degrees

$$\theta = a_1 \cos\left\{\frac{360\, n \sqrt{J_{w1}}}{\sqrt{g\, G_1\, J_{s1}}}\, x + \gamma_1\right\} \qquad (28)$$

When $x = l_1$

$$\theta = a_1 \cos\left\{\frac{360\, n \sqrt{J_{w1}}}{\sqrt{g\, G_1 J_{s1}}}\, l_1 + \gamma_1\right\} \qquad (29)$$

The expression $\dfrac{360 \sqrt{J_w}\, l}{\sqrt{g\, G\, J_s}}$ should be calculated for

each step. We will call it the angular function and denote it by the symbol Φ

$$\Phi = \frac{360 \sqrt{J_w}\, l}{\sqrt{g}\, G\, J_s} \tag{30}$$

l being the length of the particular step with which we are working.

To satisfy equation (29) graphically we then lay off from OC an angle in degrees equal to $\Phi\, n$ and draw the arc DC.

Erect a perpendicular DE. Then if OA represents the deflection at the beginning of the shaft, OE will represent the deflection at the end of the first step.

Let DE be denoted by y_1. Lay off a distance $EF = y_1^1$ so that

$$y_1^1 = y_1 \frac{\sqrt{G_1 J_{s1} J_{w1}}}{\sqrt{G_2 J_{s2} J_{w2}}} \tag{31}$$

It is now seen that OF $= a_2$ and $\angle$ FOA $= \gamma_2$ for equation (26) is thus satisfied and since

$$a_1 \sin (\beta_1 l_1 + \gamma_1) = y_1$$

and

$$a_2 \sin \gamma_2 = y_1^1$$

we have satisfied equation (25).

Lay off now from OF an angle $\Phi_2\, n$ and proceed as for the second step until the end of the shaft is reached, say on the line OJ or OJ[1]. At this end of the shaft suppose there to be attached a rotating mass of moment of inertia J_2. Then if M_e is the turning moment due to the acceleration of this rotating mass

$$M_e = -\frac{J_2}{g} \frac{\delta^2 \theta}{\delta t^2} \tag{32}$$

And

$$M_e = G_e J_{se} \frac{\delta \theta}{\delta x} \tag{33}$$

Equating (32) and (33) and proceeding in the same manner as by which we derived equation (19) we get

$$\tan\,(\beta_e\, l_e + \gamma_e) = -\frac{2\,\pi\, n\, J_2}{\sqrt{g\, G_e\, J_{se}\, J_{we}}} \tag{34}$$

The e subscripts denoting the end or last step. Equation (34) will be satisfied if

$$\beta_e\, l_e + \gamma_e = \angle\ \mathrm{AOJ} + m\,\pi$$

m being o or any integer and $\angle$ AOJ $< \pi$

and
$$\tan \angle\ \mathrm{A\,O\,J} = -\frac{2\,\pi\, n\, J_2}{\sqrt{g\, G_e\, J_{se}\, J_{we}}}$$

If nothing is attached to this end of the shaft $\beta_e\, l_e + \gamma_e$ will equal $\pi + m\,\pi$

Now for each value of the integer m some value of n will satisfy all the equations and end conditions throughout the shaft and will therefore be one of the natural periods of vibration of the shaft. When $m = 0$, n will be the period of the vibrations of primary order, for we see from the polar diagram that θ will pass through the 0 value once, that is, there will be one node. When $m =$ I we see that θ will pass through the 0 value twice and we will have vibrations of secondary order.

We wish to find, then, the values of n by which the construction in the polar diagram will cover a multiple of 180 degrees. If we are solving for primary vibrations we wish to cover 180 degrees; if for secondary 360; third order, 540 degrees, etc.

The value of n which will do this is determined by trial and error.

We will now review the actual procedure of calculation for easy reference.

(1) The shaft is divided into steps of suitable length.

(2) For each of these steps an equivalent J_w and GJ_s is calculated. In other words, we have substituted for the actual

shaft an equivalent shaft made up of steps of uniform size and loading. Note that masses attached to the extreme ends of the shaft are not considered as belonging to any step.

(3) Calculate for each step the angular function given by equation (30)

$$\Phi = \frac{360 \sqrt{J_w}\, l}{\sqrt{g G}\, J_s}$$

And the break function given by equation (31)

$$= \sqrt{G J_s J_w}$$

(4) If to the ends of the shaft rotating masses are fastened calculate for each end the tangent function given by equation (20) or (34)

$$\tan \gamma_1 \text{ or } \tan (\beta_e l_e + \gamma_e) = \frac{2 \pi n J_1}{\sqrt{g G J_s J_w}}$$

(5) Take one end of the shaft as the beginning and the other as the end and calculate for each break the ratio of the break functions, marking them on the diagram of the shaft for convenience.

That is find

$$\frac{y_1^1}{y_1}, \quad \frac{y_2^1}{y_2}, \quad \frac{y_3^1}{y_3}, \quad \text{etc.}$$

(6) Assume now some value for n, and starting at the beginning of the shaft lay off in the diagram OA equal to unity and AC equal to tan γ_1.

From OC lay off an angle DOC $= \Phi_1 n$, multiply DE in the ratio $\frac{y_1^1}{y_1}$, lay off $\angle$ FOG $= \Phi_2 n$, and proceed thus until the end of the shaft is reached.

Lay off finally the angle whose tangent has been calculated for the rotating mass at the end of the shaft.

The total angle we have now passed over will be a measure of the closeness of the first assumption for n from which a

second approximation can be made, and thus on to the required degree of closeness. Or we might plot a curve of total angle and n and find for what values of n this curve cuts the 180 or 360 degree line, whichever is required.

Once the functions have been calculated for each step the approximate diagrams can be constructed very rapidly.

It is apparent at once that a diagram that fulfills the necessary conditions, that is, that covers a multiple of 180 degrees,

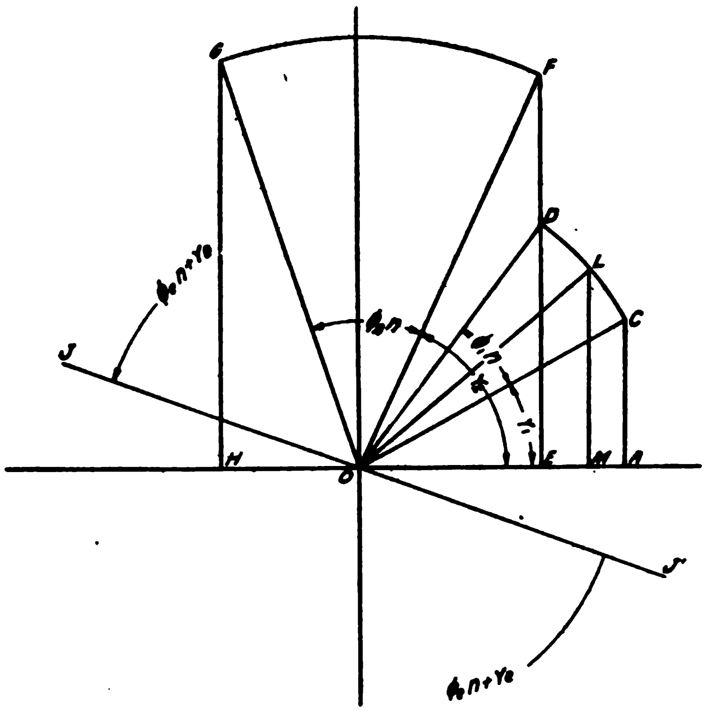


FIG. 2.

is a diagram of relative angular shaft deflections due to torsional vibration. In Fig. 2, for instance, OA is the deflection at the beginning of the shaft, OE the deflection at the end of the first step, OH at the end of the second step, etc. The

deflection at any point within the step is found by dividing the angle for the step, as COD, in the same ratio as the point divides the length of the step. To find the deflection at the middle of the first step, for instance, draw OL bisecting angle DOC. Then OM is the deflection.

In this manner the position of the nodes is instantly determined, a node being a point of zero deflection is the point on any step where its arc cuts the vertical through the origin.

We will illustrate the calculations for a simple case.

To one end of a shaft 122 feet long and 10 inches diameter, is fastened a propeller of J equal 3,550 in foot pound units, and to the other end a gear of J equal 12,500. Determine the period.

As there is but one step it will not be necessary to construct a polar diagram.

For the shaft,

$$J_s = \frac{\pi}{32} \times \frac{10^4}{12^4} = .0474$$

At the gear end

$$\tan \gamma_1 = \frac{2\pi \times 12{,}500 \times n}{\sqrt{32.16 \times 12{,}000{,}000 \times 144 \times 490} \times .0474} = .318n$$

At the propeller end

$$\tan \gamma_2 = \frac{2\pi \times 3550 \times n}{\sqrt{32.16 \times 12{,}000{,}000 \times 144 \times 490} \times .0474} = .0904n.$$

And

$$\Phi n = \frac{360 \times \sqrt{490} \times 122 \times n}{\sqrt{32.16 \times 12{,}000{,}000 \times 144}} = 4.11\, n$$

We determine then the values of γ_1, γ_2 and Φn for various values of n and tabulate

n	γ_1	γ_2	Φn	$\gamma_1 + \gamma_2 + \Phi n$
1	17.65	5.4	4.11	27.16
5	57.8	24.3	20.55	102.65
10	72.5	42.1	41.1	155.7
15	78.2	53.7	61.5	193.4
20	81.0	61.0	82.2	224.3
30	84.0	69.8	123.3	277.1
40	85.5	74.5	164.4	324.4
50	86.4	77.5	205.5	369.4

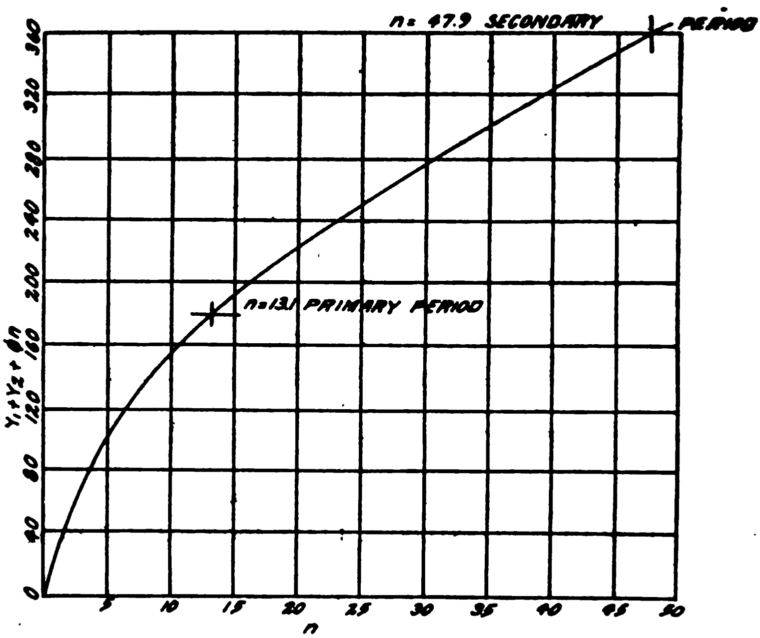
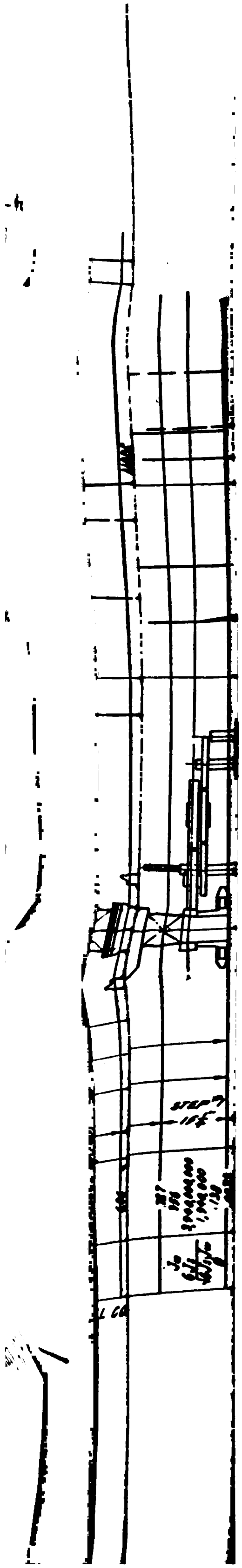


FIGURE 3

In Fig. 3, $\gamma_1 + \gamma_2 + \Phi n$ is plotted to a base of n. We see that for $\gamma_1 + \gamma_2 + \Phi n = 180$ degrees, $n = 13.1$; and that for $\gamma_1 + \gamma_2 + \Phi n = 360$ degrees, $n = 47.9$. The primary period is therefore 13.1 vibrations per second and the secondary 47.9.



To determine the position of the primary node we note that for $n = 13.1$,

$$\begin{aligned} \gamma_1 &= 76.5 \\ \gamma_2 &= 49.8 \\ \Phi n &= \underline{53.7} \\ & \quad 180.0 \end{aligned}$$

Therefore the primary node is $\frac{90-76.5}{53.7} \times 122 = 30.65$ feet from the gear end.

Also for $n = 47.9$
$$\begin{aligned} \gamma_1 &= 86.1 \\ \gamma_2 &= 76.9 \\ \Phi n &= \underline{197.0} \\ & \quad 360.0 \end{aligned}$$

Therefore the first secondary node is $\frac{3.9}{197} \times 122 = 2.42$ feet from the gear end.

And the second secondary node is $\frac{183.9}{197} \times 122 = 113.9$ feet from the gear end.

The calculations have been made to determine the period of the shafting of U. S. Submarines *S-4—S-13* and are illustrated here. The shafting is shown in figure 4. There are three cases to be investigated. First, the charging condition when the engine is driving the dynamo, the after clutch being disconnected. Second, the surface condition, when the engine drives the propeller. Third, the submerged condition when the motors drive the propeller, the forward clutch being disconnected.

The main shaft was divided into ten steps as shown and for each of these J_w and GJ_s calculated. In finding J_w in way of the cranks one-half of the reciprocating weight was used, and in finding the moment of inertia of the propeller 25 per

cent was added to the calculated amount to allow for the entrained water.

In calculating GJ_s, G was taken at 11,790,000 * * *. See JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS, Vol. XIV, Page 731. To determine the value of GJ_s in way of the cranks a torsion experiment was made upon the actual engine shaft in its bearings. The value determined by this experiment for GJ_s was 4,620,000,000. The value of GJ_s in way of the crank bearings is 3,940,000,000. The shaft with cranks is therefore 17.3 per cent stiffer than the plain shaft. In way of the clutches the value of GJ_s was decreased somewhat to allow for the effects of play.

On each step are marked the values of J_w, GJ_s, $\sqrt{GJ_s J_w}$ and Φ and on the breaks the ratios from right to left of $\sqrt{GJ_s J_w}$.

The closing polar diagrams for primary and secondary vibrations are shown in figures 5 to 9. In the charging condition for primary vibrations the period was found to be 24.7. As there are 8 cylinders and therefore 4 impulses per revolution, the critical speed would be $24.7 \times 60/4 = 361$ revolutions per minute. In operation, vibrations of great violence were felt at speeds around 350 revolutions per minute, being most violent at speeds between 340 and 360 revolutions per minute. The period of the secondary vibrations for this condition was 38.8, corresponding to a critical speed of 584 revolutions per minute, which is higher than any at which the engine was run.

In the surface condition the primary period works out at 15.8, corresponding to a critical speed of 237 revolutions per minute. The period of the secondary vibrations is 24.7, corresponding to a critical speed of 370 revolutions per minute. In operation violent vibrations were felt at speeds around 365 revolutions per minute.

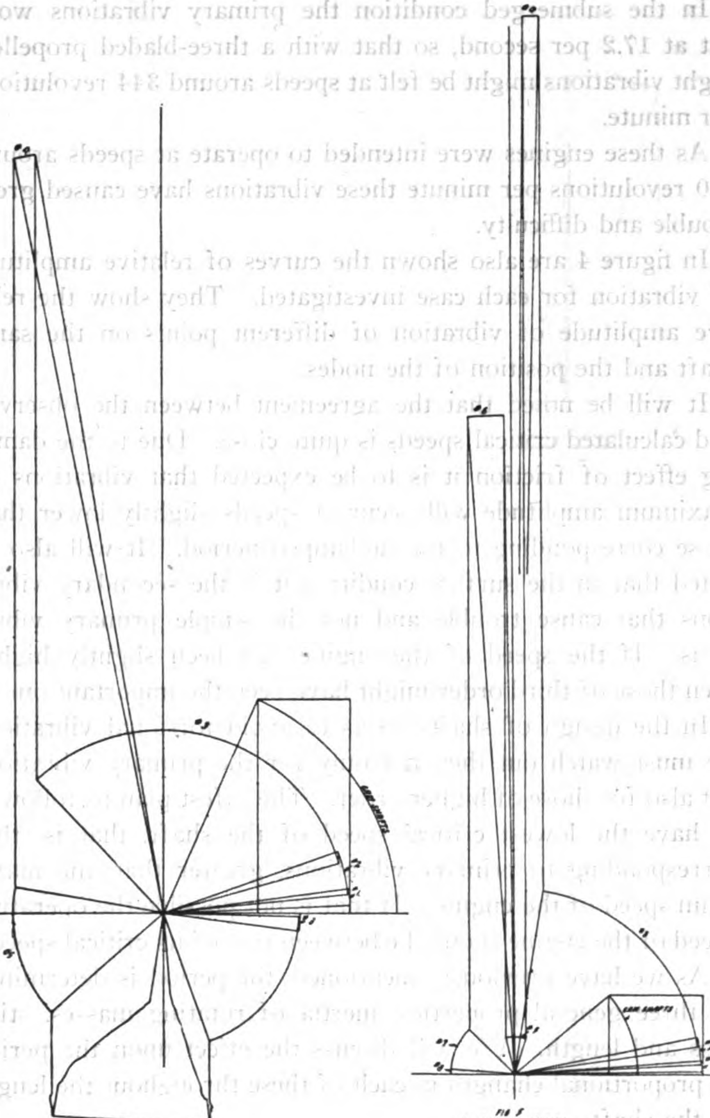


FIG. 4
CHARGING CONDITION
SECONDARY VIBRATIONS
n=38.8

FIG 5
CHARGING CONDITION
PRIMARY VIBRATIONS
n=241

U.S. SUBMARINES S4-S13
TORSIONAL VIBRATION ANALYSIS OF SHAFTING
POLAR DIAGRAMS

In the submerged condition the primary vibrations work out at 17.2 per second, so that with a three-bladed propeller, slight vibrations might be felt at speeds around 344 revolutions per minute.

As these engines were intended to operate at speeds around 350 revolutions per minute these vibrations have caused great trouble and difficulty.

In figure 4 are also shown the curves of relative amplitude of vibration for each case investigated. They show the relative amplitude of vibration of different points on the same shaft and the position of the nodes.

It will be noted that the agreement between the observed and calculated critical speeds is quite close. Due to the damping effect of friction it is to be expected that vibrations of maximum amplitude will occur at speeds slightly lower than those corresponding to the undamped period. It will also be noted that in the surface condition it is the secondary vibrations that cause trouble and not the simple primary vibrations. If the speed of the engine had been slightly higher even those of third order might have been the important ones.

In the design of shafts so as to avoid torsional vibrations we must watch out then not only for the primary vibrations but also for those of higher order. The safest plan to follow is to have the lowest critical speed of the shaft, that is, that corresponding to primary vibrations, greater than the maximum speed of the engine. If that is not possible the operating speed of the engine should lie between two of its critical speeds.

As we have previously mentioned, the period is determined by three general properties, inertia of rotating masses, stiffness and length. We will discuss the effect upon the period of proportional changes in each of these throughout the length of the shaft.

It is apparent without proof that if we change any property of the shaft in the same proportion thoughout its length that the polar diagram will remain unchanged.

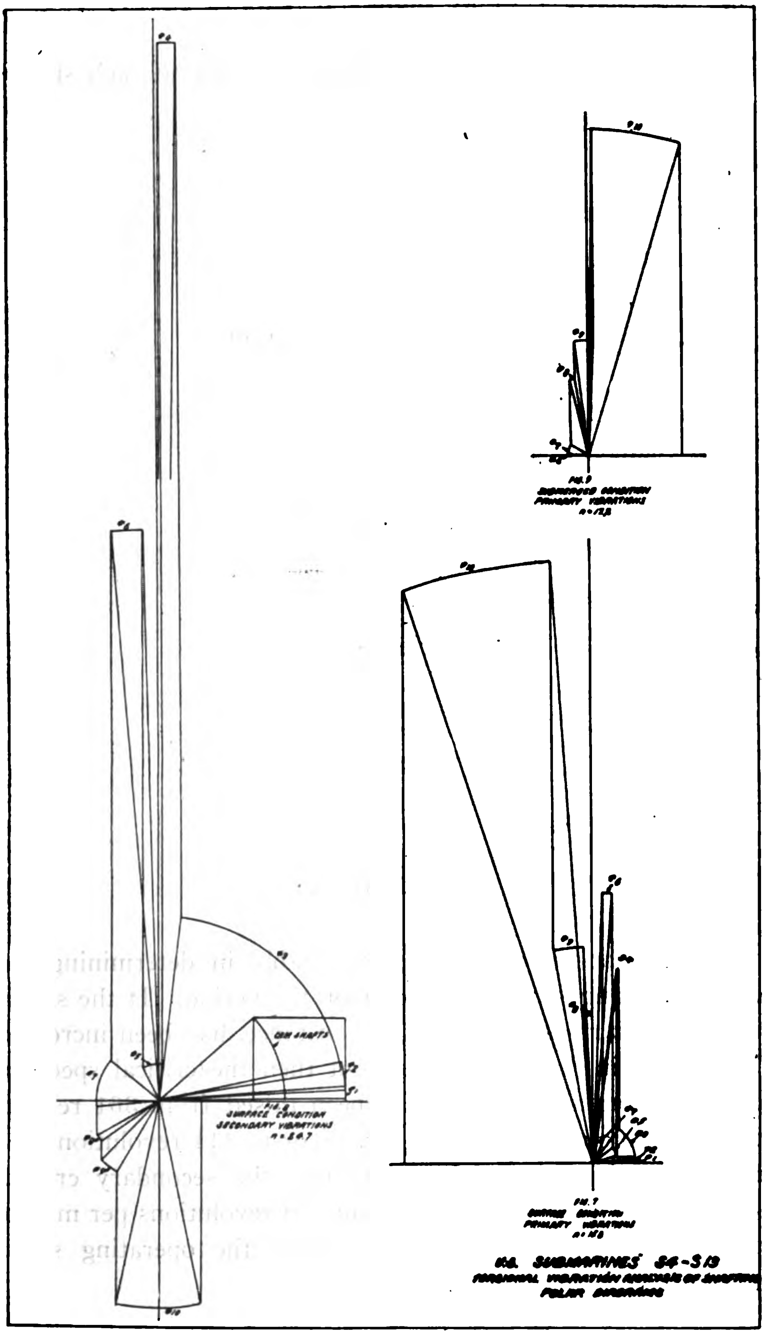


FIG. 8
SURFACE CONDITION
SECONDARY VIBRATIONS
n = 24.7

FIG. 9
SUBMERGED CONDITION
PRIMARY VIBRATIONS
n = 12.5

FIG. 7
SURFACE CONDITION
PRIMARY VIBRATIONS

U.S. SUBMARINES S4-S13
TORSIONAL VIBRATION ANALYSIS OF SHAFTING
POLAR DIAGRAMS

By equation (29) the angle corresponding to each step is equal to

$$\Phi\, n = \frac{360 \sqrt{J_W}\, l\, n}{\sqrt{g\, G\, J_S}}.$$

So that we can write

$$\frac{360 \sqrt{J_W}\, l\, n}{\sqrt{g\, G\, J_S}} = \text{Constant}$$

From which it follows that
For changes in J_w only

$$n \text{ varies as } \frac{1}{\sqrt{J_W}}$$

For changes in GJ_s only

$$n \text{ varies as } \sqrt{G\, J_S}$$

For changes in l only

$$n \text{ varies as } \frac{1}{l}$$

For similar engines

$$n \text{ varies as } \frac{1}{l}.$$

These relations assist us considerably in determining the effect of changes in dimension upon the period. If the shafting of submarines *S-4—S-13*, for instance, had been increased 7 per cent in diameter throughout then the critical speed in charging condition would have been raised from 361 revolutions per minute to $361 \times (1.07)^2$, or to 414 revolutions per minute; and in the surface condition the secondary critical speed would have been raised from 370 revolutions per minute to 424 revolutions per minute. Thus the operating speed would have been avoided.

It is, of course, not necessary to change the shafting throughout. A change in any part would alter the period, and it would be often possible by relatively slight changes in proposed dimensions, when those changes are made judiciously, to radically alter the critical speed, and thus avoid difficulties.

The treatment in such special cases as gears and turbines in mesh with the main shaft, cam shafts driven by it, elastic couplings, etc., has not been taken up here, but it is believed that any one who has carefully followed the demonstration can readily devise such special treatment as the need may arise.

U. S. S. *RAMAPO*

(EMERGENCY FLEET CORPORATION OIL TANKER NO. 1655).

DESCRIPTION AND TRIALS.

BY HENDERSON B. GREGORY, ASSOCIATE.

The *Ramapo* is the second of eight oil tankers contracted for, under date of October 10, 1918, with the Newport News Shipbuilding & Dry Dock Co. and the Navy Department, on behalf of the U. S. Shipping Board Emergency Fleet Corporation.

Under the terms of the contract the price of the hull and machinery was based upon an estimated cost of $2,200,000.00, plus or minus the cost of authorized changes, the contractors to receive a fixed profit of $220,000.00; this profit to be augmented, in the event of the actual cost being less than the estimated cost or authorized revision thereof, by one-third the amount saved.

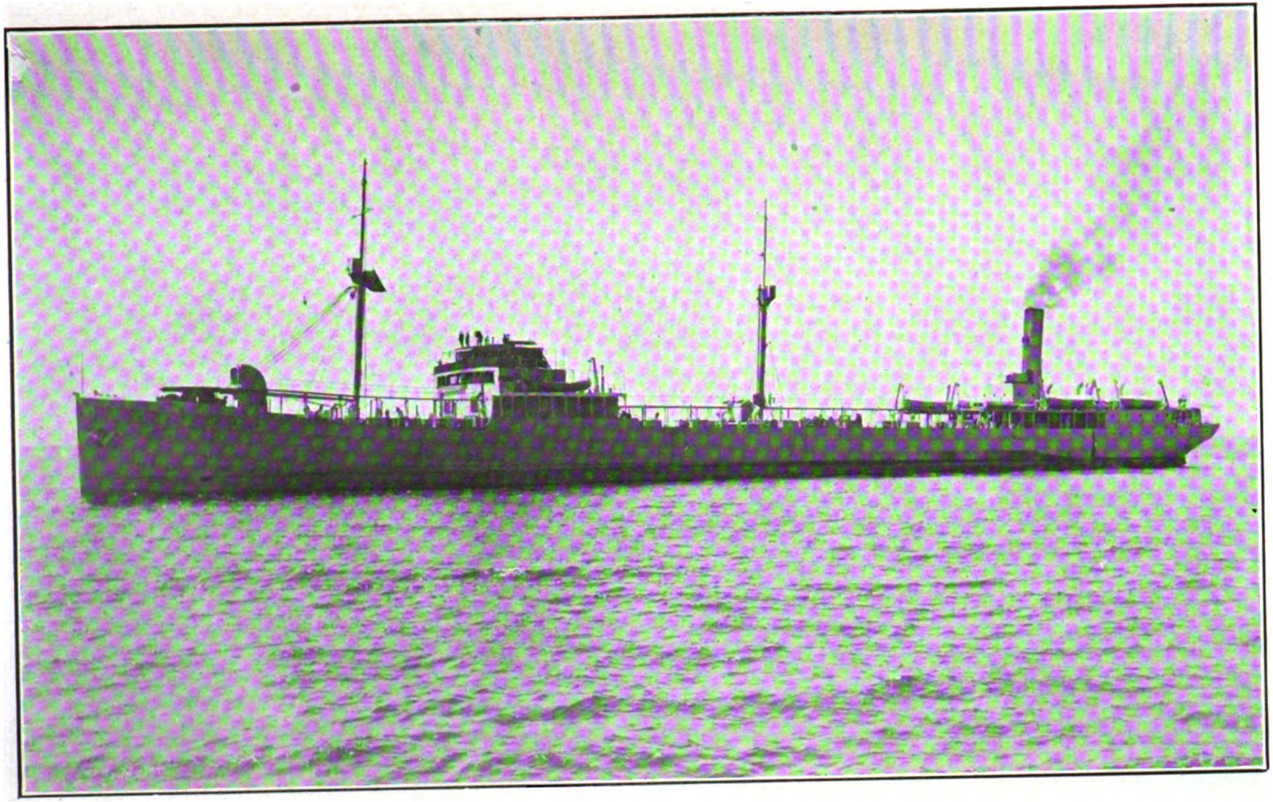
The hull and machinery are in general accord with the Rules of the American Bureau of Shipping for highest classification, and the U. S. Steamboat Inspection Rules for seagoing vessels. The Navy Department also inspected all work.

The keel was laid January 16, 1919, and vessel was launched September 11, 1919. Thirty-four days after the launching the vessel proceeded to sea for her trials.

GENERAL DESCRIPTION OF HULL.

The hull is of the following principal dimensions:

Length over all, feet and inches	477–10
between perpendiculars, feet and inches	463–3
on load-water line, feet and inches	462–10



U. S. S. "RAMAPO."

Beam, extreme, feet and inches	60–2⅝
on load-water line, feet and inches	60–0
moulded, feet and inches	60–0
Mean designed draught, feet and inches	26–2
Displacement on designed draught, tons	16,785
per inch on load-water line, tons	57.9
Area of immersed midship section at designed draught, square feet	1,551
of load-water line plane at designed draught, square feet.	24,318
Wetted surface at designed draught, square feet	46,000
Coefficient of fineness, block	0.812
midship section	0.988
load-water line	0.873

The vessel is a single screw, steel steamship, with straight stem, elliptical stern and with flat sheer amidships. It is rigged with two masts and one smoke pipe. It is of the shelter-deck type, with complete steel shelter, upper and main decks, except that the upper and main decks are omitted in way of the oil tank expansion trunks and the machinery space. The longitudinal system of framing and scantlings is followed throughout.

The propelling machinery is located in the after end of the hold, and a screen bulkhead separates the engine and boiler rooms. Immediately abaft the machinery compartment is the after peak tank. Forward the boiler room and separated therefrom by a cofferdam, are the fuel oil bunker tanks, of 63,000 cubic feet capacity.

Reserve feed water is carried in tanks under the machinery space, and there is a ballast tank beneath the fuel oil bunkers.

The amidship hold is divided into nine tanks, of about 450,000 cubic feet total capacity, for carrying oil in bulk. The tanks are provided with center-line bulkhead extending to the shelter deck, dividing them into port and starboard compartments, and there are expansion trunks between the main and shelter decks for the full length of the tanks. The wing spaces, between the main and upper decks in way of the main oil tanks, are divided into five compartments, port and starboard, fitted

up as summer oil tanks. The total capacity of these tanks is 43,000 cubic feet. There is a pump room between oil tanks Nos. 5 and 6, and a cofferdam forward of tank No. 1, between tanks Nos. 3 and 4, and between tank No. 9 and the fuel oil bunker tanks.

The following are located in the forward hold: a pump room, cargo space, fuel oil or ballast tanks, chain locks and the fore peak tank.

Aft on the main deck are located the cold storage compartments, fresh water tanks and storage space. Forward are cargo space and stores.

The upper deck aft is fitted as crew's quarters. The sick bay is located on this deck on the starboard side of the engine hatch. Forward are crew's quarters, prison, carpenter shop, lamp room, emergency generator room, paint room and windlass engine.

Aft on the shelter deck is a deck house for the steering gear, with gun platform on top of the deck house. A similar gun platform is provided in the bow over the windlass. No guns are mounted.

The crew's mess is located in a deck house abreast of the engine hatch, port side, on the shelter deck, with spacious galley on center line abaft the engine hatch. The corresponding deck house on the starboard side of the engine hatch contains the general mess pantry, stores, laundry, and engineer's office.

The radio room is on top of the after deck house.

Forward is a double tier deck house containing the officers' quarters. It is raised 42 inches clear of the shelter deck. On top of this deck house are the wheel house, chart room and bridges.

The tops of all deck houses and gun platforms are connected by fore-and-aft bridges.

LIFE BOATS, ETC.

The following life boats and life rafts are carried in skid-beams on top of the after deck houses, except the whale boats, which are carried on the forward deck house:

No.	*Type.*	*Capacity each.*
4	26-ft. Metallic life boats	50 persons
2	24-ft. Whale boats	23 persons
1	24-ft. Motor boat	
1	Racine life raft	24 persons
2	Life buoy rafts	25 persons

All boats are swung in Norton sheath screw davits.

DECK MACHINERY.

Windlass.—A 14-inch × 12-inch, double engine, Hyde anchor windlass, with four gipsy heads, is located on the shelter deck forward; the engine is below on the upper deck. Patent anchors are carried, which are stowed in the hawse pipes.

Deck Winches.—Three deck winches are provided; they are all of the Hyde Windlass Company's design. The bow and midship winches have two 8-inch × 10-inch steam cylinders; the stern winch, which is of greater power, has two 9-inch × 14-inch steam cylinders. All winches have two gipsy heads.

Steering Engine.—The steeering engine and gear are located in a deck house aft on the shelter deck. It is of the Brown Steam Tiller type, as manufactured by the Hyde Windlass Co. The steam cylinders are 9-inch × 12-inch.

Ammunition Hoists.—Four small Lidgerwood ammunition hoists are provided, two for each gun. They are driven by G. E. Co. motors. The ammunition is carried in small magazines in the forward and after holds.

CARGO OIL PIPING.

The cargo oil suction system consists of two 11-inch steel pipe lines, leading aft from the pump room, one on each side of

the center line bulkhead, for draining tanks 6 to 9, inclusive. Four similar lines lead forward, two for tanks 1, 2 and 3, and two for tanks 4 and 5. Each tank connects to its main through a 9-inch angle valve fitted with bell mouth carried down to near the bottom of the tank. A master gate valve is fitted in the suction main at the transverse bulkhead nearest the pump room of each tank served. All valves are operated from the shelter deck. Valves are arranged that the pumps may draw from any tank and discharge to any other tank.

A 16-inch gate valve, operated from the shelter deck, is fitted on the center-line bulkhead between each pair of main cargo tanks.

The summer tanks are drained by 6-inch pipes. Four 6-inch filling pipes, two port and two starboard, terminating on the shelter deck, are provided for these tanks; they connect to the 6-inch drain mains.

Two 11-inch risers are fitted forward, from the suction mains to the shelter deck, for filling tanks 1, 2 and 3, and two similar risers aft for tanks 6 to 9, inclusive. Tanks 4 and 5 are filled through connections on the 11-inch discharge mains at the pump room hatch, port and starboard. Each filling connection terminates at the deck with an 11-inch gate valve and 11-inch × 8-inch × 8-inch Y-casting with flanged ends. Eight-inch × 6-inch flanged reducers are provided to suit 6-inch U. S. Navy standard oil hose.

Cargo oil is discharged through two 11-inch connections in the pump room, leading to the 11-inch cross-connection on the shelter deck and the discharge mains on the upper deck, port and starboard. These mains are 11 inches diameter, extending aft along the upper deck from the pump room hatch for about 112 feet, where they rise to the shelter deck and are carried aft full size for a short distance and then reduced to 6 inches, continuing to the after end of the deck house abreast of the engine room. Deck discharge connections are taken off these mains. Forward of the pump room hatch the mains are

8 inches diameter for a distance of about 60 feet, where they branch into two 6-inch connections on each side of the vessel, one rising abruptly to the shelter deck and the other extending forward about 75 feet before rising to the shelter deck. All deck connections are flanged to suit 6-inch U. S. Navy standard oil hose.

The cargo tanks are flooded through two 10-inch sea valves in the pump room, either direct or through the pumps, and are pumped overboard via the shelter deck cross-connection. Four portable sections of 8-inch pipe of sufficient length to extend from the discharge outlets to the ship's side are provided for pumping overboard.

The forward fuel oil or ballast tanks are served by a fuel oil transfer pump located in the forward pump room. These tanks are provided with 6-inch deck filling connections, 5-inch suctions and a 4-inch discharge to the fuel oil transfer line and deck, the latter provided with a portable section of piping extending to the ship's side for overboard discharge.

All oil tanks are provided with vents, steam fire-extinguishing connections and steam heating coils.

DESCRIPTION OF MACHINERY.

The main propelling machinery is located in a common compartment in the after hold.

Main Engine.—The main engine is of the vertical, inverted, direct-acting, quadruple-expansion type. It has a stroke of 51 inches, and the following cylinder diameters: H.P., 24 inches; 1st I.P., 35 inches; 2nd I.P., 51 inches; and L.P., 75 inches. The engine turns to starboard when running ahead, and is designed to develop about 2,800 I.H.P., with steam at 220 pounds' pressure at the H.P. valve chest, when driving the vessel at the designed speed of 10.5 knots.

The engine is of conservative design, with cylinders of the usual construction; all fitted with working liners except the L.P. The cylinders are supported on cast iron columns of the

box type at the front and the inverted Y type at the back. Each column carries a crosshead guide. The bed plate is in four sections. It is of cast iron of box section, flanged and securely bolted together and to the engine foundation.

All reciprocating parts are of forged steel. Adjustable cast iron shoes, for working on the guides, are secured to each crosshead. The H.P. and 1st I.P. pistons are of cast iron; those for the 2nd I.P. and L.P. cylinders are cast steel. All pistons are fitted with cast iron junk rings. U. S. metallic packing is used for all piston rods and valve stems.

The oil system is through wick feed from brass manifold oil boxes.

The valve gear is of the Stephenson link type. All valves are fitted with balance pistons, except the H.P. piston valve. The valves and their settings are given in Table 1.

TABLE I.—VALVE SETTINGS.

	H. P.		1st I. P.		2d I. P.		L. P.	
Number	One		One		One		One	
Type	Piston		Piston		Slide		Slide	
Diameter, inches	11 and 11½		18 15/16 and 19		....		....	
Travel, inches	8		8		8		9	
Takes steam	Inside		Outside		Outside		Outside	
	Top	Bot.	Top	Bot.	Top	Bot.	Top	Bot.
Width of port, inches	2½	2½	2½	2½	4¾×43	4¾×43	6×67	6×67
Steam opening, linear, inches	1 13/16	2 1/16	1 29/32	2 1/32	2 7/64	2 7/32	2 13/32	2 17/32
Steam opening, area, sq. ins.	54.25	54.65	87.04	92.75	181.4	190.8	330.81	339.18
Exhaust steam, linear, inches	2½	2½	2½	2½	4¾	4¾	6	6
Exhaust steam, area, sq. ins.	70	66.25	114	114	204.25	204.25	402	402
Steam lap, inches	2 1/16	1 5/16	2 3/32	1 31/32	1 57/64	1 29/32	2 1/32	1 31/32
Exhaust lap, inches	−1 1/32	−21/32	−1/8	0	−1/2	0	−23/64	+23/64
Steam lead, linear, inches	3/8	1/2	11/32	15/32	5/8	3/4	7/8	11/16
Cut-off, decimal of stroke	.73	.652	.728	.652	.742	.658	.743	.657

Connecting rod, length between centers, 115 inches. Ratio to crank 4.51.

Reversing Gear.—The reversing engine is secured to the bottom of the port column of the 2nd I.P. cylinder. It is of the ram type, with cylinder 14 inches diameter by 24 inches stroke. It is controlled from the working platform through the reversing lever and a floating lever attached to the reverse shaft. The reverse arms are slotted and fitted with screw blocks for independent linking up.

Turning Gear.—For jacking the main engine a vertical, single, reciprocating engine is provided. It is located on the after end of the main engine, port side. The steam cylinder is 8 inches diameter by 6 inches stroke.

Shafting and Bearings.—All shafting is forged steel of solid section, with flanged couplings forged on and secured together by forged steel bolts.

The crank shaft is in four sections of the built-up type. The four cranks are at equal angles, the sequence being H.P., 1st I.P., L.P., and 2nd I.P.

The propeller shaft is covered, within the stern tube, with composition sleeves ⅞-inch thick at the bearings and ½-inch thick between bearings. The sleeves are secured in the usual manner.

The shafting is of the following principal dimensions:

Crank shaft, length, feet and inches	29–8
diameter, inches	14¼
pins, length, inches	13
diameter, inches	14¾
Thrust shaft, length, feet and inches	11–6
diameter, inches	14¼
collars, number	10
thickness, inches	2¼
diameter, inches	23½
space between, inches	5
bearing surface, square inches	2742.6
Line shaft, length, feet and inches	13–0
diameter, inches	13½
Propeller shaft, length, feet and inches	17–3¼
diameter, inches	15

The thrust bearing is of the usual adjustable horse-shoe marine type.

The shaft bearings are of the following characteristics:

Crank shaft bearings:	
number	8
length, inches (1) 13½, (6) 15 and (1)	16½
diameter, inches	14¼
bearing surface	White metal
Thrust shaft bearings:	
number	2
length, inches	14¼
diameter, inches	14¼
bearing surface	White metal
Thrust shoes, number	10
effective surface, square inches	1920
Line shaft bearings:	
number	2
length, inches	20
diameter, inches	13½
bearing surface	White metal
Stern tube bearings:	
Forward, length, inches	24
diameter, inches	16 13/16
bearing surface	Lignum vitae
After, length, inches	60
diameter, inches	16¾
bearing surface	Lignum vitae

Propeller.—The propeller is a right-hand true screw. It has four adjustable, detachable blades of manganese bronze. The hub and cap are cast iron. Each blade is secured to the hub by seven 3-inch steel studs and composition nuts. The hub has a taper fit on the propeller shaft, is securely keyed, and held on by a steel nut locked with a key.

PROPELLER DATA.

Diameter, feet and inches	17–6
Pitch as set, feet and inches	18–0

Pitch adjustable from 16 feet 8 inches to 19 feet 4 inches.

Pitch ratio	1.022
Projected area, square feet	85.8
Helicoidal area, square feet	101.4
Disc area, square feet	240.5
P.A. ÷ D.A.	0.3565
Height of lower tip of blade above keel, inches	13⅜
Immersion of upper tip of blade at load draught, inches	92⅞

Main Condenser.—There is one independent, cylindrical, double-flow, surface condenser located on the port side of the main engine. The shell and heads are of cast iron, tube sheets of rolled Muntz metal 1¼ inches thick, and seamless-drawn Admiralty metal tubes No. 16 B.W.G. thick, ¾-inch outside diameter and 8 feet 10⅝ inches long. The tubes are packed with corset lacing and brass ferrules. There are 2,471 tubes, giving a total cooling surface of 4,200 square feet.

Main Circulating Pump.—The circulating water is supplied by a double-suction, 14-inch, centrifugal pump, driven by a direct connected reciprocating engine having a cylinder 8 inches in diameter by 10-inch stroke. Two engines are provided, one as standby.

Main Air Pump.—The air pump is attached to the main engine and located abreast the L.P. cylinder, port side. It is of the vertical, double-acting, Edwards type, and has a 26-inch diameter cylinder with a 24-inch stroke.

Attached Bilge Pump.—There is one vertical bilge pump of the plunger type attached to the main air pump. It is 3½ inches diameter by 24-inch stroke.

Attached Sanitary Pump.—A vertical, sanitary pump, of same type and size as the bilge pump, is also attached to the main air pump.

Auxiliary Condenser.—A small auxiliary condenser of the Wheeler surface type, combined with horizontal air and circulating pump, is located in the engine room outboard of the main condenser. It has 577 tubes, ¾-inch outside diameter, No. 16 B.W.G. thick and 6 feet 5⅛ inches long, giving a cooling surface of 705 square feet.

TABLE II.—INDEPENDENT RECIPROCATING PUMPS.

No.	Service.	Size, Type and Make.	Location.
2	Main feed	12 in. × 8 in. × 24 in. vertical, single, double-acting, "Worthington,"	Engine room
1	Auxiliary feed	12 in. × 8 in. × 24 in. vertical, single, double-acting, "Worthington."	Engine room
1	Donkey feed	8 in. × 5 in. × 12 in., horizontal, simplex, double-acting, "Worthington."	Engine room
1	Sanitary	6 in. × 5¾ in. × 6 in., horizontal, duplex, double-acting, "Worthington."	Engine room
1	Auxiliary air and circulating	12 in. × 14 in. × 14 in. × 12 in., horizontal, simplex, double-acting, combined, "Worthington."	Engine room
1	Fresh water	4½ in. × 2¾ in. × 4 in., horizontal, duplex, double-acting.	Engine room
1	Evaporator feed	4½ in. × 2¾ in. × 4 in., horizontal, duplex, double-acting.	Engine room
2	Ballast, fire and bilge	8 in. × 8½ in. × 12 in., horizontal, duplex, double-acting, "Worthington."	1 in engine room 1 in forward pump room
2	Fuel oil service	5¼ in. × 3½ in. × 5 in., horizontal, duplex, double-acting.	Boiler room
2	Fuel oil transfer	10 in. × 6 in. × 10 in., horizontal, duplex, double-acting, "National Transit Pump & Machine Co."	1 in boiler room 1 in forward pump room
2	Cargo oil	18 in. × 14 in. × 24 in., horizontal, duplex, double-acting, "National Transit Pump & Machine Co."	Pump room

Feed Water Heater.—In the engine room, outboard and above the feed pumps, is a No. 12 Reilly, type C, multicoil feed-water heater. Between the feed pumps and the feed heater is a grease extractor. Both the feed heater and the grease extractor can be by-passed.

Hotwell Tank.—One hotwell tank of about 500 gallons total capacity is installed. It is fitted with a float and chronometer valve gear for automatically regulating the feed pumps.

Independent Reciprocating Pumps.—Table II gives the principal data for all independent reciprocating pumps.

Boilers.—There are three main, horizontal, single-ended, Scotch boilers and one horizontal, single-ended, Scotch donkey boiler, all fitted with separate combustion chambers, and arranged to burn fuel oil. The main boilers are placed abreast just forward of the main engine, with fire room forward. The donkey boiler is located in the boiler hatch at the upper deck level.

BOILER DATA.

	Main (each).	Donkey.
Working pressure, pounds	220	180
Test pressure, pounds	330	270
Diameter, external, feet and inches	14–7 3/16	11–1
Length, over heads, feet and inches	11–6	10–10½
Furnaces, number	3	2
type	Morrison,	Suspension
diameter, internal, inches	43	39
length, external, feet and inches	8–1 13/16	7–9⅜
Heating surface, square feet	2345	1223
Tubes, material	Charcoal iron,	lap welded
number, ordinary	250	132
stay	100	56
outside diameter, inches	2¼	2¼
thickness, ordinary, B.W.G	No. 10	No. 10.
stay, inches	5/16	5/16
length, ordinary, inches	93½	89
stay, inches	94¼	89¾
Safety valve, type	Twin,	Ashton
size	3	2½

Uptakes.—Uptakes of usual design for Howden, heated air, forced draft, connect the main boiler tube nests to the smoke pipe. For heating the air, 789 3-inch by No. 12 B.W.G. tubes, presenting a total heating surface of 2,169 square feet, are installed vertically in the uptakes. The gases of combustion pass through the tubes and the forced draft air outside.

The donkey boiler uptake connects into the smoke pipe immediately below the top of the boiler hatch.

Dampers are fitted in all uptakes.

Smoke Pipe.—There is one double casing smoke pipe, 8 feet 1 inch outside diameter and 80 feet high above the center of the lower furnaces. The clear cross-section area through the smoke pipe is 41.29 square feet.

The donkey boiler smoke pipe is inside the main smoke pipe and extends to the top. It is of single casing construction 33 inches inside diameter.

Forced Draft.—Howden, closed ash pit, heated air, forced draft is used for the main boilers. The air is supplied by one vertical, engine-driven, Sturtevant Type VS7, double-inlet. blower, with 110-inch diameter fan. The fan is provided with duplicate single engines, one being a standby. Each engine has a cylinder 7 inches diameter by 6 inches stroke, and is capable of driving the fan at 390 revolutions per minute. The unit is located in the engine room, starboard side, forward. The air is drawn from the engine room and delivered, through sheet-iron ducts of rectangular section, to the heater boxes in the uptakes, thence to the ash pits.

Fire Room Hoist.—A hoist of the steam ram type is fitted in the starboard ventilator to the fire room.

Fuel Oil System.—The Dahl system of mechanical oil-fuel burning is installed for both the main and donkey boilers. The apparatus comprises one oil burner per furnace, three oil heaters, two oil-fuel service and two oil-fuel transfer pumps, together with the necessary strainers, piping, etc. The service pumps are arranged to draw from all bunker tanks,

including summer tanks No. 5 and cofferdam No. 3, which are fitted for carrying bunker oil-fuel, and they discharge to the burners through the oil-fuel heaters or bypass same. All bunker tanks are fitted with high and low suctions.

Steam Piping.—The main steam pipe is of seamless drawn steel. The feeders from the boilers are 6 inches each, uniting into an 8-inch pipe leading to the main engine throttle valve. A 4½-inch auxiliary steam connection is taken off each main boiler, forming a main with branches to the engine room and deck, from which sub-branches are taken as required. The donkey boiler has a 4-inch connection to the auxiliary steam line. A 2½-inch independent steam line is led from the donkey boiler and No. 1 main boiler to the generator sets in the engine room; they also take steam from the auxiliary steam line in the engine room.

Auxiliary Exhaust Piping.—The exhaust steam from all auxiliaries is collected in a common main connecting to the main and auxiliary condensers, and the feed water heater. A back pressure valve is fitted at each condenser connection to maintain the necessary pressure on the line for the feed water heater. There is also a connection from the auxiliary exhaust line to the 2nd I.P. receiver.

Feed System.—Four feed pumps are provided as listed in Table II. All feed pumps take their suction from the hotwell tank. The main and auxiliary feed pumps are arranged to feed the main boiler direct or through the feed water heater. The donkey feed pumps feeds the donkey boiler direct or the main boilers via the auxiliary feed line. The feed stop and check valves are at the rear end of the boilers and are controlled from the engine room. One No. 13½ and one No. 10½ Metropolitan injector are also connected up to the feed lines.

Evaporating and Distilling Plant.—The plant consists of two Reilly multicoil evaporators, each having a capacity of 35 tons of fresh water per 24 hours, and two Reilly multicoil, vertical

distillers, each of 6,000 gallons' capacity per 24 hours, together with the necessary pumps as listed in Table II. The evaporators are located in the engine room, at the working level, starboard side, aft, with the distillers above at high elevation in the engine hatch.

Refrigerating Plant.—The ice machine room is starboard of the engine room, and it is entered from the middle grating. The outfit includes a 2-ton Kroeschell, vertical, double acting, CO_2 compressor driven by a direct-connected steam engine, oil separator, CO_2 receiver and condenser, together with the usual appurtenances.

The refrigerating rooms are abaft the engine room on the main deck.

Lighting Plant.—There are two G. E. Co. direct current, 20-KW, horizontal, compound-wound, Type MP6-20-400, generators, each driven by a 9-inch × 7-inch vertical, single, direct-connected, steam engine. They are located on a flat in the after end of the engine room. The switchboard is just abaft the generators. Current is supplied at 110 volts.

An emergency generator-set is also provided. It is located on the upper deck forward and consists of a Westinghouse, direct current, 12.5 kilo-watt, 125 volts, generator, designed for 600 revolutions per minute, and driven by a 25 horsepower, four-cylinder, Clifton Motor Works kerosene engine.

Workshop.—A small workshop is provided starboard of the engine room, at the level of the middle grating. It is equipped with the following tools and equipment:

1 16-inch Steptoe shaper.
1 Buffalo Forge Co. hand drill.
1 12-inch Blount, double-energy grinder.
1 Grindstone.
1 16-inch Davis lathe.
1 20-inch Fairbanks upright drill.
1 Anvil.
1 Portable forge.

1 Work bench.

2 Vises.

Outfit of hand tools.

All tools are belt driven from overhead shaft and pulleys. The power is supplied by a 7 H.P., G. E. Co., direct-current motor, designed for 1,150 revolutions per minute.

TRIALS.

The contract required:

(*a*) A trial over a measured mile in the deepest water available and at the highest speed obtainable, not less than three runs over the mile to be made, and the mean speed and revolutions for these runs carefully ascertained.

(*b*) A full speed trial of four hours' duration in deep water at the highest speed attainable (not less than 10½ knots), to be determined by the average revolutions of the main shaft according to the measured mile trial. During this trial all the auxiliaries necessary for the service of the vessel, including dynamos for efficient lighting, shall be in operation. The fuel oil consumption on this trial to be measured and not to exceed the equivalent of 325 pounds per knot at 10½ knots.

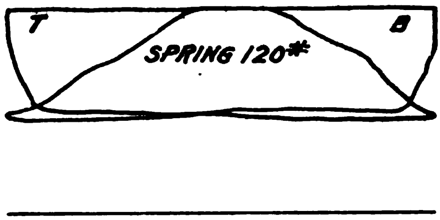
(*c*) Turning and backing trials to test the engines and steering gear.

The *Ramapo* sailed from Newport News for her trials on the morning of October 15, 1919, and proceeded to the Delaware Breakwater, where the standardization runs were made over the measured mile.

The four-hour official full-power trial was conducted on the run up the coast to the Delaware Breakwater. At the builders' request, the oil and water consumption, the latter not required by the contract, were measured as follows: During the first three hours of the official trial the total oil consumption and the water consumption of the main engine and auxiliaries was measured. Then followed two hours at full power measuring

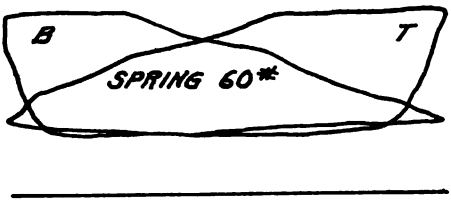
U.S.S. RAMAPO
FULL POWER TRIAL INDICATOR CARDS OF
SET TAKEN AT 2:45 P.M., OCT. 15, 1919.
R.P.M. 74.9 – I.H.P. 2839

H.P. CYL.



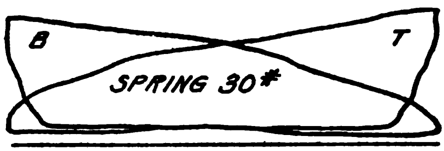
M.E.P. 84#
M.R.P. 8.36#
I.H.P. 710
ST. CHEST 222#
CUTOFF 67%

1ST I.P. CYL.



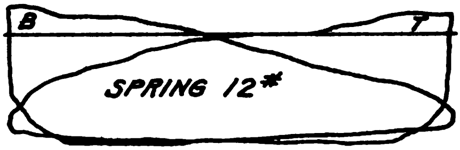
M.E.P. 39.3#
M.R.P. 8.46#
I.H.P. 719
REC'R. 99#
CUTOFF 68.7%

2ND I.P. CYL.



M.E.P. 18#
M.R.P. 8.3#
I.H.P. 705
REC'R. 37#
CUTOFF 68.8%

L.P. CYL.



M.E.P. 8.3#
I.H.P. 705
REC'R. 7#
VACUUM 27.5"
CUTOFF 68%

PLATE I.

Table III - U.S.S. RAMAPO - Trial Data.

	Full Power Trials.			Reduced Power Trials.	
	4 Hours Official	3 Hours Measuring Oil, & Water M.E.& Auxs.	2 Hours Measuring Oil, & Water M.E. only.	4 Hours Measuring Oil, & Water M.E.& Auxs.	2 Hours Measuring Oil, & Water M.E. only.
Date of trial	Oct.15,1919	Oct.15,1919	Oct.15,1919	Oct.16,1919	Oct.16,1919
Time	11:45 A.M.	11:45 A.M.	3:15 P.M.	2:15 P.M.	6:30 P.M.
at 10.5 knots	---	273	---	---	---
guaranteed	---	325	---	---	---
below guarantee	---	52	---	---	---
hour per I.H.P. main engines	---	1.1	1.113	1.107 **	1.079
sq. ft. of H.S.	---	0.462	0.477		0.393
Miscellaneous data:					
Per cent. CO_2	9.8	10	9.35	10.5	9.7
Number of boilers in use	3	3	3	3 for 1½ hrs. 2 " 2½ "	2
" " burners in use	9	9	9	4 to 6	4 to 6
Main engine linked up, inches	---	1-0-0-1½	1-0-½-1½	5-2-2-3	5-2-2-3

* Approximate.
** One boiler cut out during run.
*** All water and oil consumptions corrected for extra pumps used for measuring water and oil.

the total oil consumption and the water consumption of the main engine only, other conditions remaining unchanged. This was done during the last half hour of the official trial and continued an hour and a half thereafter. The data obtained is given in Table III.

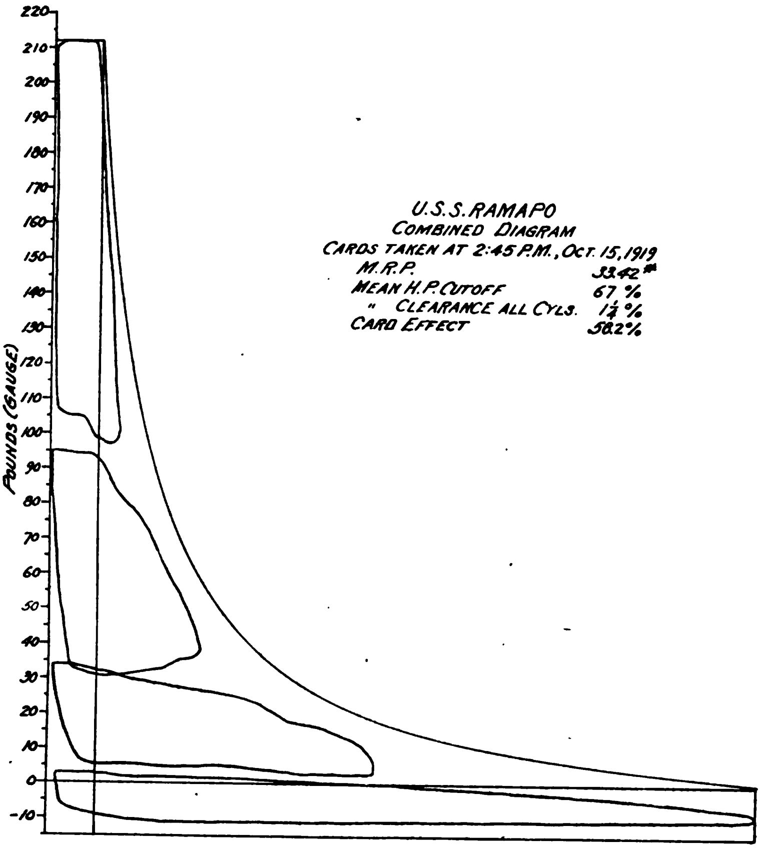


PLATE II.

Plate I shows a set of indicator cards taken during the official trial and Plate II shows the combined card of same set.

Following these trials the steering and backing tests were conducted. While making 70 revolutions per minute ahead

the vessel was swung through a figure eight course. Time required to put rudder hardover right from center was 12 seconds, estimated diameter of turning circle 650 yards. Time required to put rudder hardover left from hardover right was 22 seconds, estimated diameter of turning circle, 550 yards.

Upon completion of the figure eight the engine was reversed and the backing and astern steering test was run with the following results:

Time for vessel to become dead in water from full speed ahead with rudder amidships......3 minutes, 56 seconds.
Headway reached..............................575 yards.
Time to put rudder hardover right from center....17 seconds.
Time to put rudder hardover left from hardover right ....23 seconds.
Time to put rudder center from hardover left..... 8 seconds.
Time for ship to become dead in water from full speed astern with rudder amidships.....2 minutes, 56 seconds.
Stern way reached............................200 yards.

The standardization trial was run on the morning of October 16, 1919, and from the data obtained the curves (Plate III) were plotted. From the revolutions per minute and speed curve it was ascertained that 69.14 revolutions per minute of the propeller would give the designed speed of 10.5 knots.

During the homeward trip the builders conducted two private runs; one of four hours' duration, with the main engines linked up to make about 60 revolutions per minute, and measuring the total oil consumption and the water consumption of the main engine and auxiliaries, and the other under similar conditions for two hours, measuring the total oil consumption and the water consumption of the main engine only. The data from these trials is also given in Table III.

The data given in Table III for the official four-hour trial is that taken during the trial by the builders and should not differ from the official figures by any appreciable amount.

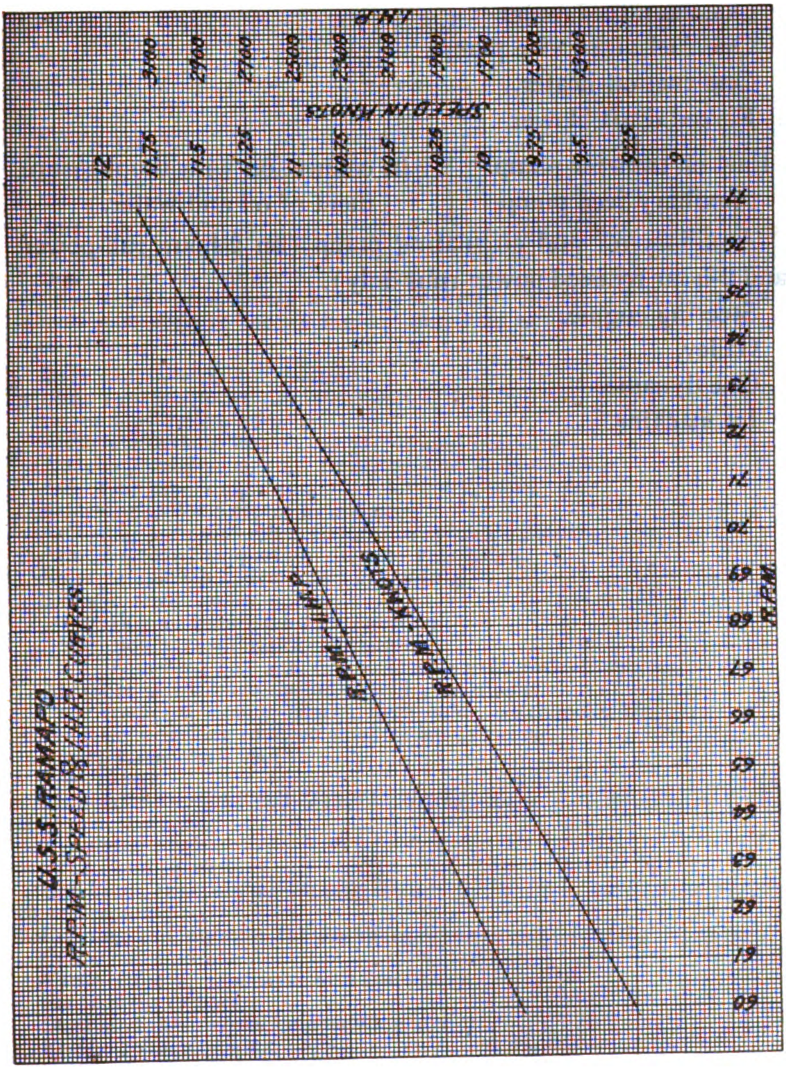

U.S.S. RAMAPO
R.P.M.-SPEED & I.H.P. CURVES
R.P.M.-I.H.P.
R.P.M.-KNOTS
SPEED IN KNOTS
I.H.P.
R.P.M.
12
11.75
11.5
11.25
11
10.75
10.5
10.25
10
9.75
9.5
9.25
9
3100
2900
2700
2500
2300
2100
1900
1700
1500
1300
60
61
62
63
64
65
66
67
68
69
70
71
72
73
74
75
76
77

PLATE III.

The following auxiliaries were in use on all the trials:

Main circulating pump.

One main feed pump.

One fuel oil service pump.

One fuel oil heater.

Forced draft blower.

Fuel oil transfer pump.
Auxiliary feed pump. } (In connection with the measurement of oil and water.)

Auxiliary air and circulating pump (on 2-hour full-power and 2-hour reduced power runs only).

Steering engine.

Galley.

One dynamo.

Ice machine.

NOTES.

PARTS OF VESSEL TO WHICH APPLICATION OF ELECTRIC WELDING HAS BEEN CONSIDERED AND APPROVED BY AMERICAN BUREAU OF SHIPPING AND LLOYDS.

Although the classification societies are proceeding very cautiously in approving the application of electric welding to new ship construction, because of the lack of a reliable system of testing the quality and strength of the welds, nevertheless both the American Bureau of Shipping and Lloyds now approve the welding by electricity of the following parts of steel vessels:

Deck rail stanchions to plating.
Clips for detachable rail stanchions.
Continuous railing rods (joints).
Attaching deck collars (L rings) around ventilators.
Attaching deck collars (L rings) around smoke stack.
Attaching cape rings around smoke stack, pipes, etc.
Attaching galley fixtures to plating.
Attaching bath and other fixtures in officers' quarters.
Attaching cowl supporting rings to ventilators.
Bulwark rail top splicing and end fitting.
Skylights over galley.
(*a*) Engine room stairs and gratings.
(*b*) Boiler room stairs and gratings.
Attaching (*a*) and (*b*) to plating grab rods on casing.
All stairs and ladders, including rail attachments.
Door frames to casing, hinges, catch holds, coachhooks, etc.
Clips for attaching interior wood finish to casing.
Entire screen bulkhead.
Coal chutes.
Butts of watertight and oiltight boundary bars on bulkheads or floors in double bottom.
Ventilator cowls.
Stacks and uptakes.
Bulkheads (that are not structural parts of the ship), partition bulkheads in accommodation.
Framing and supports for engine and boiler room flooring or gratings.
Cargo batten cleats.
Tanks (that are not structural parts).
Shaft alley escapes.
Steel skylights over accommodation spaces.
Engine room skylights.
Grab rods on exterior and interior of deck houses.
Deck houses not covering unprotected openings through weather decks.
Reinforcing and protecting angles around manholes.
Joints of watertight angle collars at frames in way of watertight flats.—"International Marine Engineering."

THE EVOLUTION OF THE DESTROYER.

Remarkable Development of Present-Day Destroyer from Early Type Torpedo Boat—Rapid Advances in Boiler and Turbine Construction.

By Commander S. M. Robinson, U. S. N.

Progress in marine engineering has been very rapid during the past thirty years, but it is believed that the development of the torpedo boat has been more rapid than that of any other type of vessel. Undoubtedly the main reason for this rapid progress is that it takes a very short time, comparatively, to build a destroyer, so that the engineer can test his ideas and make several advances in the time required to build one battleship. It is certainly a far cry from the 22.5-knot *Cushing* to the 36.88-knot *Dent*. The *Cushing* was our first torpedo boat and the *Dent* is one of our latest destroyers. The following table and accompanying photographs give a comparison between the two boats:

	Cushing.	*Dent.*
Length overall.........	137 feet 6 inches	314 feet 4½ inches
Beam, extreme.........	15 feet ½ inch	30 feet 1½ inches
Draft	4 feet 6 inches	9 feet 0¾ inch
Displacement	91.34 tons	1,159 tons
Fuel	33 tons coal	288 tons oil
Speed	22.5 knots	36.88 knots
Horsepower	1,720 (indicated)	28,190 (shaft)
Revolutions per minute.	370	467.5
Number of screws.....	2	2
Cruising radius at 10 knots' speed..........	990 knots	at 15 knots, 4,960 knots
Boilers	2 Thornycroft	4 White-Forster
Total heating surface..	4,750 square feet	27,000 square feet
Total grate surface.....	76.6	
Ratio of heating surface to grate surface......	62 to 1	

The *Cushing* was our first torpedo boat and was designed and built in 1890 by Nathaniel Herreshoff at the Herreshoff Works, Bristol, R. I. Inasmuch as she represents our maiden effort in this line of marine engineering, a brief description of her machinery may be of interest.

MACHINERY OF THE "CUSHING."

She was equipped with twin screws, each driven by a five-cylinder, quadruple expansion engine. The cylinders were 11½, 16, 22½, 22½ and 22½ inches in diameter, respectively. The stroke was 15 inches. The engine supports were a distinct advance in engineering for that day and consisted of 1½-inch steel rods, braced diagonally, and forming the cap bolts of the main bearings; the latter were secured to a bed-plate consisting of a single sheet of ¾-inch wrought iron with openings cut for the cranks. The bed-plate was secured in a fore-and-aft direction to keelsons and was also supported under the main bearings of the high-pressure and after low-pressure cylinders, the other main bearings being entirely without support; but, owing to the extreme care taken in balancing, the engines ran without vibration.

The boat was equipped with two Thornycroft boilers each having 38.3 square feet of grate surface and 2,375 square feet of heating surface. The boilers also represented an advance in engineering, as the longitudinal seams in the drums were welded instead of riveted. The boilers were designed for a pressure of 250 pounds per square inch.

For some time after the trials of the *Cushing,* the advance in torpedo boat engineering was limited by the progress in the design of propellers and the art of balancing engines. Like many other points of engineering, the screw propeller was not as well understood at that time as it is today, and many failures on trials were recorded due solely to improper propeller design. Also the higher speed of ship desired required higher engine speeds than had been the custom, with the result that vibration was so bad in many cases as to make it impossible to run at full power. However, both of these problems were successfully solved and the art of engine balancing, in particular, was carried to a high degree of success. This line of development reached its climax with the *Stewart* class, which were built in 1902.

THE LAST OF THE RECIPROCATING ENGINE BOATS.

The following table gives the general characteristics of these boats:

Length	245 feet
Beam	23 feet ¼ inch
Draft	6 feet 1¼ inches
Displacement	444 tons
Speed	29.3 knots
Indicated horsepower	8,000
Revolutions per minute	330
Number of screws	2
Fuel, coal	180 tons
Boilers	4 Thornycroft
Total heating surface	17,770 square feet
Total grate surface	237 square feet
Ratio of heating surface to grate surface	56.4 to 1
Radius of action at 12 knots	2,160 knots

These were the last of the reciprocating engine boats. We see that the speed has increased from 22.5 knots to 29.3 knots, the boiler pressure from 250 to 300 pounds, the indicated horsepower from 1,720 to 8,000, and the high-pressure cylinder from 11½ inches to 23 inches diameter. There is no doubt but what with the present-day methods of lubrication and other improvements of design, such as superheat, etc., destroyers of higher speeds could be built using reciprocating engines, but the improvement would not be great and the destroyer had just about reached its limitation as to speed.

Up to this time torpedo boats and destroyers had been regarded as coast and harbor defense vessels and long-distance cruising had not been carried out by them. The ordinary method of operation was to make short runs at high speed from a base, but in 1904 two of these boats, the *Preble* and *Paul Jones,* were sent to Panama to act as despatch vessels, and later in the year a flotilla of destroyers was sent from the Atlantic coast to the Asiatic station, via the Mediterranean. Both cruises proved conclusively that the destroyer was a reliable sea-going vessel and had a cruising radius that compared favorably with other types of ships. The fuel economy at low speeds was the greatest surprise of all. At that time so little was known regarding the fuel consumption at cruising speeds that

when the cruise from San Francisco to Panama was first projected arrangements were made for coaling at San Diego and every few hundred miles from there to Panama. Great was the surprise when it was found that the trip could easily be made with only one stop and without any stop by taking a small deck-load of coal.

ADOPTION OF THE TURBINE FOR PROPULSION.

The next step in destroyer design was the change to turbines for propulsion, and with this came the use of forced feed for lubrication. The first vessels to have this type of machinery were the *Smith* class, which were built in 1909. The following table gives the characteristics of these boats:

Length	293 feet 10½ inches
Beam	26 feet 4½ inches
Draft	8 feet ½ inch
Displacement	716 tons
Speed	28.35 knots
Shaft horsepower	9,946
Revolutions per minute	724
Number of screws	3
Boilers	4 Mosher
Total heating surface	18,003 square feet
Total grate surface	368.5 square feet
Ratio of heating surface to grate surface	48.86 to 1
Fuel, coal	304 tons
Cruising radius at 16 knots	2,800 knots

With previous destroyers fitted with reciprocating engines it had been difficult to obtain machinery that would be reliable for high-speed runs for any considerable period of time. The difficulty was so great that most of the early destroyers were required to run full-speed trials of one-hour duration only. As soon as turbines were adopted, the trial requirements were immediately raised to four hours. The destroyer, being a high-speed boat, requires high speed of revolutions of the screw, and this condition, while ideal for turbines, is a very severe test for reciprocating engines. There is no doubt but what the destroyer was the main factor in developing turbines for marine purposes, as it presented a case where turbines were fairly well suited for the purpose without any great change from conditions on shore. This was not the case with battleships and other vessels having slow-speed screws, and the progress of turbines for marine propulsion would doubtless have been very slow if it had not been for the destroyers.

BLOWER TROUBLES ELIMINATED BY TURBINE DRIVE.

Up to this time the forced draft blowers of destroyers had all been driven by light, high-speed, reciprocating engines and they were subject to the same troubles as were the main reciprocating engines, except that the troubles were exaggerated in the smaller engines. It was only natural, therefore, to adopt the turbine for driving the blowers as well as the propellers. It is no exaggeration to say that this change was almost as great an improvement as the adoption of the turbine for propulsion, as more breakdowns and failures on trials were due to blower trouble than to all other causes.

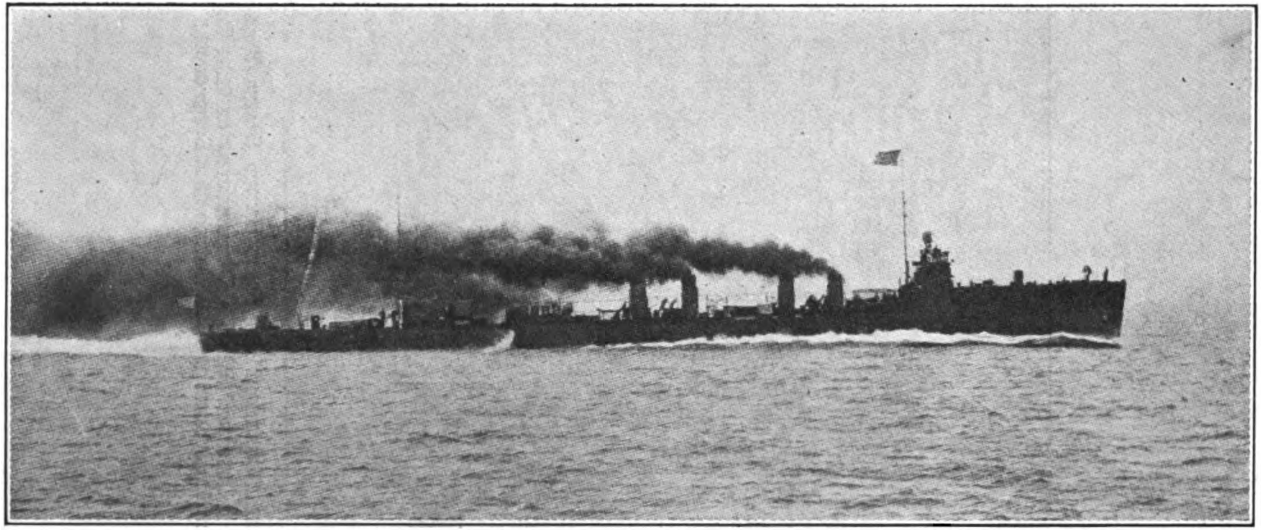
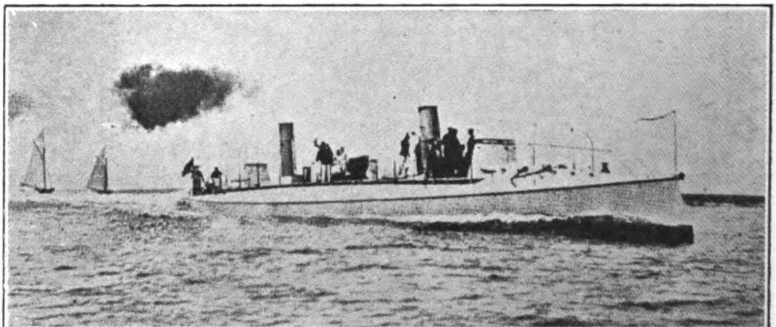


Fig. 1.—"Smith" Class Destroyer. These Vessels, Built in 1909, Were the First American Turbine-Driven Destroyers.



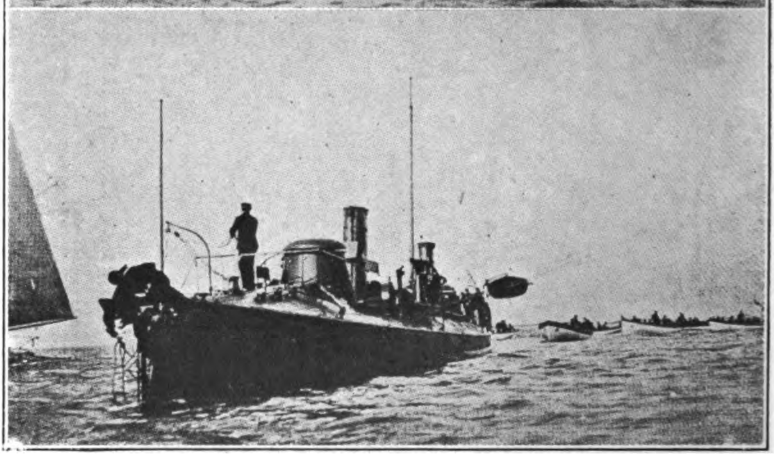
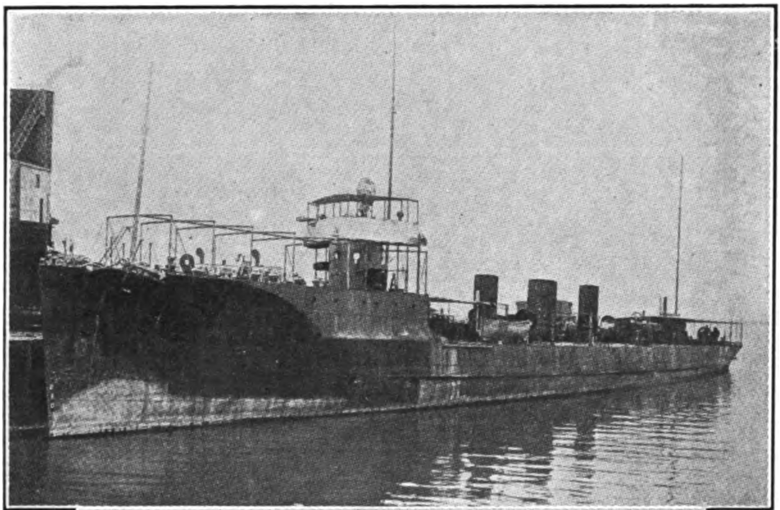


Fig. 2.—The "Cushing," the First United States Torpedo Boat.
Fig. 3.—The "Cushing," With a Displacement of 91.34 Tons and an Indicated Horsepower of 1,720, Developed a Speed of 22.5 Knots.



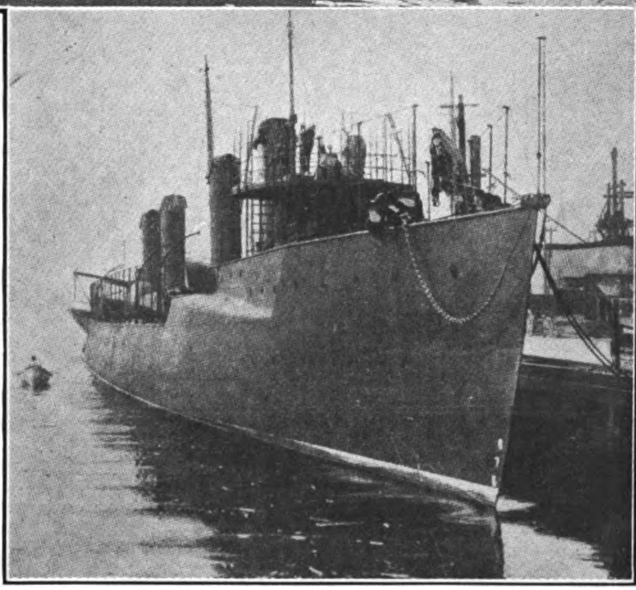


FIG. 4.—DESTROYER "ROE," BUILT AT NEWPORT NEWS IN 1910; TRIAL SPEED, 29.60 KNOTS.

FIG. 5.—"CHAUNCEY" CLASS DESTROYER (1900-1902).

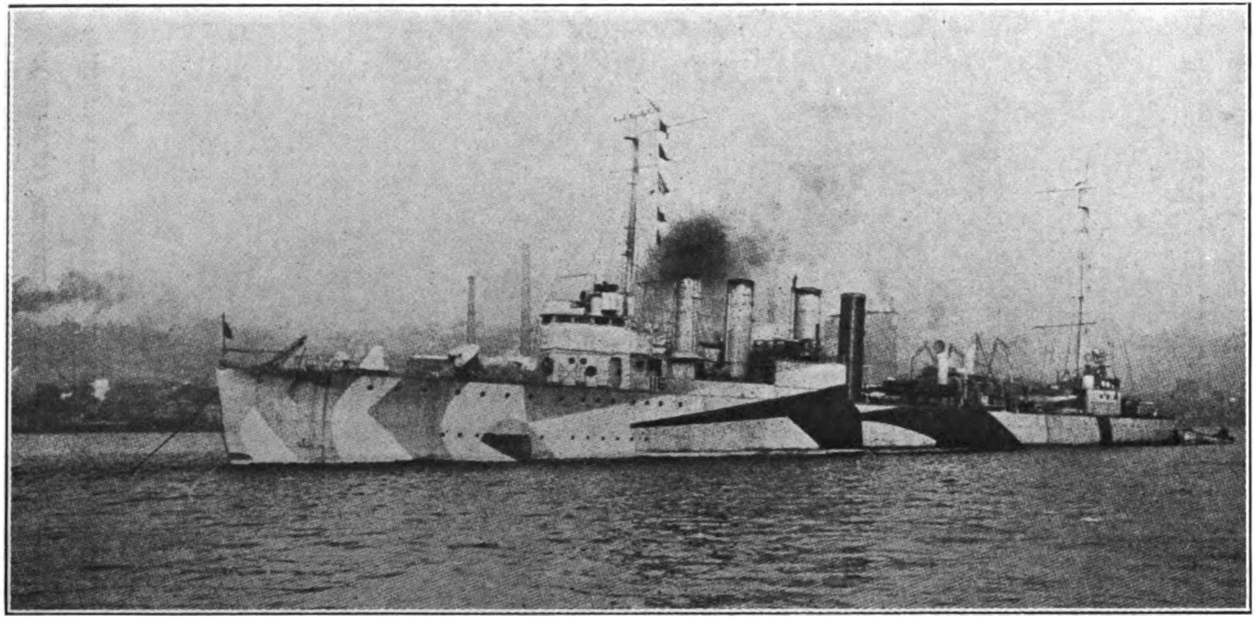


FIG. 19.—U. S. S. "DENT," LATEST TYPE AMERICAN DESTROYER; DISPLACEMENT, 1,159 TONS; SHAFT HORSEPOWER, 28,190; SPEED, 36.88 KNOTS.

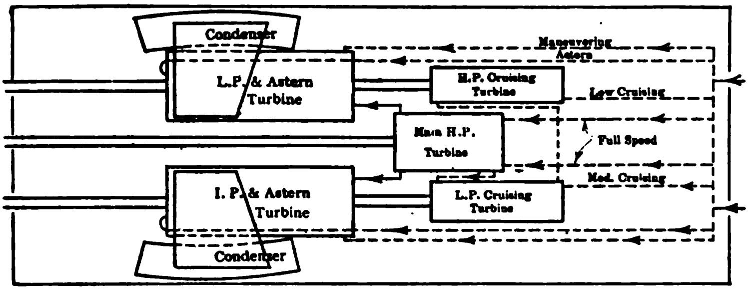


Fig. 6.—Parsons Turbines: *Smith* Class (Twenty Vessels)

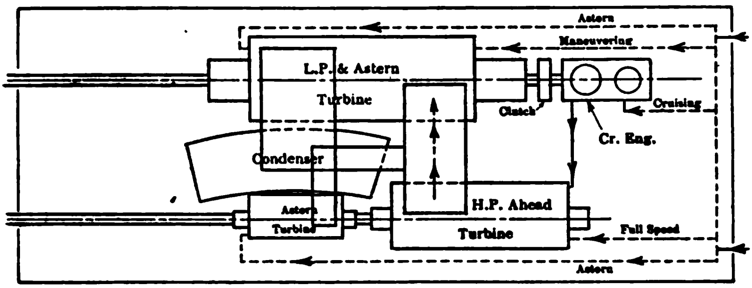


Fig. 8.—Parsons Turbines: *Cassin, Cummings, McDougall, Ericsson*

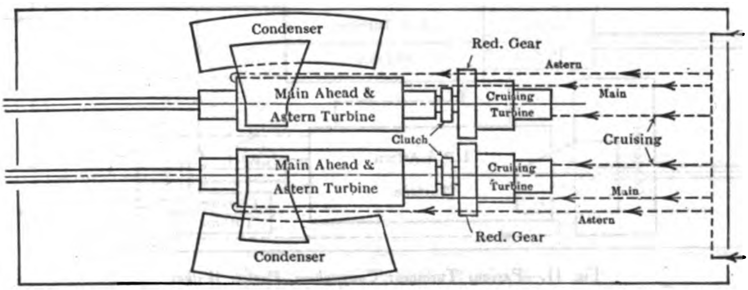


Fig. 10.—Curtis Turbines: *Cushing, Tucker, Sampson, Rowan*

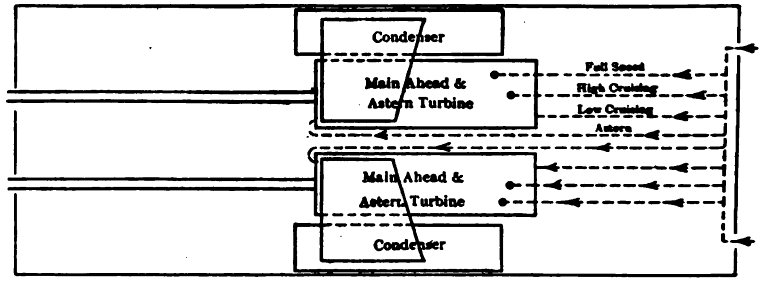


Fig. 7.—Curtis Turbines: *Perkins, Sterett, Walke.* Zoelly Turbines: *Warrington*

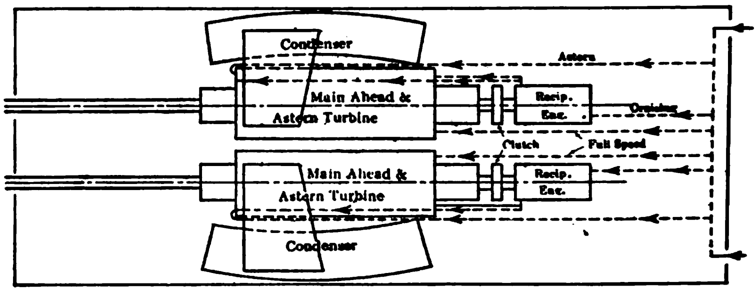


Fig. 9.—Curtis Turbines: *Downes, Duncan.* Cramp's Turbines: *Aylwin* Class (Seven Vessels)

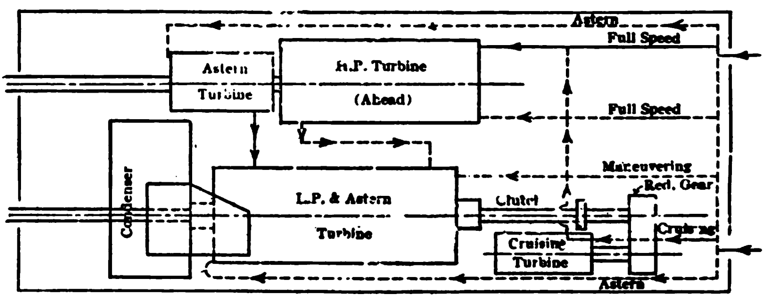


Fig. 11.—Parsons Turbines: *Conyngham, Porter, Wilkes*

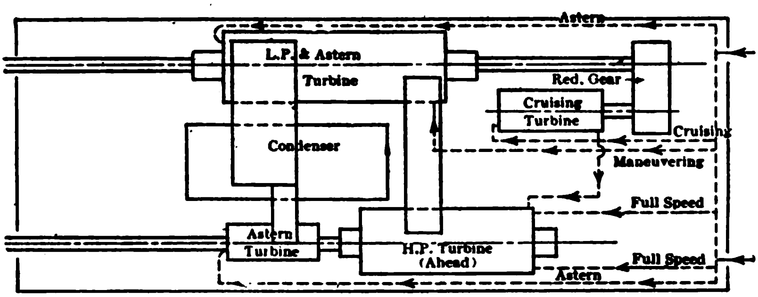


Fig. 12.—Parsons Turbines: *Jacob Jones, Wainwright, Davis, Allen, Shaw*

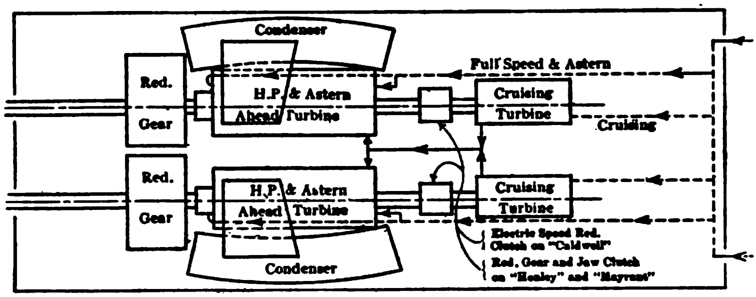


Fig. 14.—G. E. Curtis Turbines: *Caldwell.* Westinghouse Turbines: *Mayrant, Henley*

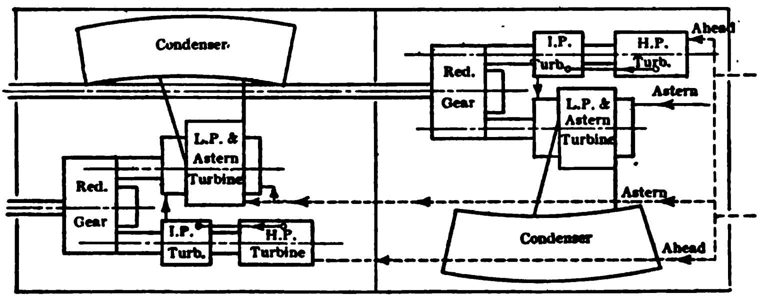


Fig. 16.—Curtis Turbines: 91 Vessels

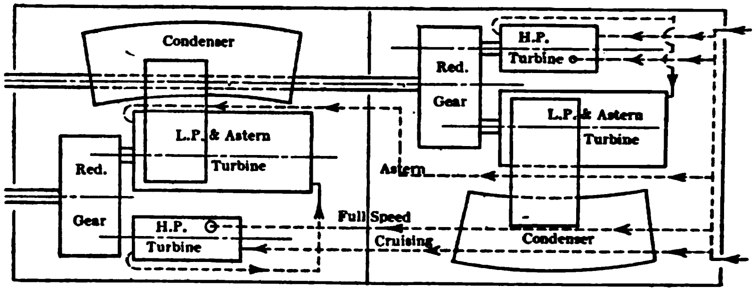


Fig. 13.—Parsons Turbines: *Wadsworth* Class (89 Vessels). Westinghouse Turbines: *Wadsworth* Class (40 Vessels)

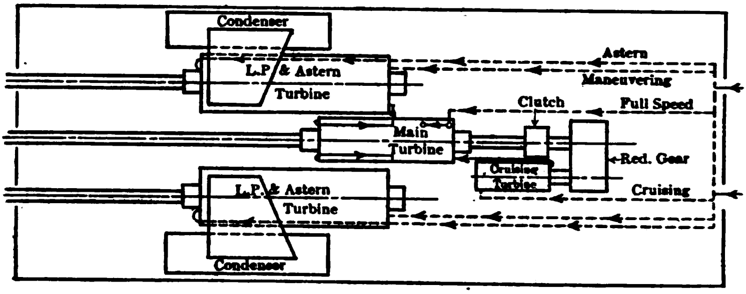


Fig. 15.—Parsons Turbines: *Conner, Stockton*

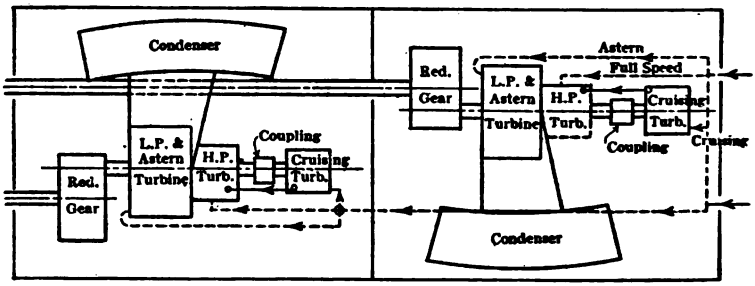


Fig. 17.—G. E. Curtis Turbines: 46 Vessels

The first turbine-driven destroyers solved the problem of reliable high-speed machinery, but there was very little saving in weight, owing to the complications that were necessary to obtain reasonable economy at low speeds. In the fight between the turbine and reciprocating engine for supremacy in the field of marine propulsion it has always been the comparatively poor economy of the turbine at low speeds that has held it back. The accompanying sketches (Figs. 6-18), giving the machinery arrangements of all our turbine-driven destroyers from the *Smith* and *Lamson* up to the present time, show the progressive steps that have been taken in the evolution of the turbine for destroyer propulsion, and it will be seen that it has been a constant struggle to simplify the machinery and keep down the weights and at the same time provide a reasonable cruising radius at the slow speeds. We have had all combinations of direct-connected cruising turbines, direct-connected cruising reciprocating engines, and geared cruising turbines.

OIL FUEL DISPLACES COAL.

The next advance in destroyer engineering was the substitution of oil fuel for coal. The advantages of this fuel were greater for destroyers than for other vessels, so it was adopted for this class of vessel at the earliest possible moment. The ability to run without making smoke, to

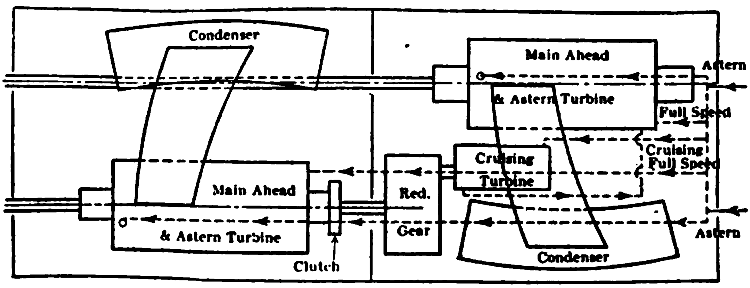


Fig. 18.—Curtis Turbines: 11 Vessels

maintain full power till the fuel is exhausted and to go quickly from low speed to full speed, all are ideal conditions for a destroyer. When the increased facility of refueling (including the ability to refuel at sea) and the increased cruising radius are added to the above advantages, it becomes apparent that the destroyer did not really fulfill its requirements until it was given oil as a fuel.

The *Paulding* class comprised the first oil-burning destroyers and they were built in 1910. The following table gives their principal characteristics:

Length	293 feet 10½ inches
Beam	26 feet 4½ inches
Draft	8 feet 1¾ inches
Displacement	711 tons
Speed	32.8 knots
Shaft horsepower	17,393
Revolutions per minute	903
Number of screws	3
Boilers	4 Normand

Total heating surface	19,321 square feet
Fuel oil	241 tons
Cruising radius at 16 knots	3,000 miles

While the propeller speeds of the destroyer were such as to allow better turbines than in the case of other ships, the speeds were not high enough for the best turbine practice, and in the attempts to improve this condition the propeller speeds used were considerably higher than should have been to get the best propeller efficiency. It was therefore natural to turn to the reduction gear as soon as it became sufficiently developed for the purpose. Here again the destroyer was the pioneer and led the way in the use of the mechanical reduction gear. This was natural, since the speed reduction was not nearly so great as with other types of ships (being only about 5 to 1), and therefore was a very much easier problem to solve; also the advantages gained were much greater than with other types of ships, owing to the great advantages of weight saving in this type of vessel.

REDUCTION GEARS APPLIED TO MAIN PROPELLING UNITS.

The *Wadsworth*, built in 1915, was the first vessel to be fitted with reduction gear for the main unit. The following table gives the general characteristics of the vessel:

Length	310 feet
Beam	29 feet 8½ inches
Draft	9 feet 4½ inches
Displacement	1,050 tons
Speed	30.7 knots
Shaft horsepower	16,100
Revolutions per minute	460
Number of screws	2
Boilers	4 Normand
Total heating surface	21,500 square feet
Fuel oil	326 tons
Cruising radius at 16 knots	5,640 knots

The arrangement of the machinery of the *Wadsworth* is shown in Fig. 13, and it will be noted that it is very simple, consisting of a high-pressure and a low-pressure turbine driving pinions meshing with the same gear, there being only two gears and two main shafts. The *Wadsworth's* machinery contains several novel features. The gear is a single helical gear and is arranged to take part of the propeller thrust direct and transmit it to the turbines, where it is balanced by steam pressure; the remainder of the thrust is taken by a thrust bearing which is of the pivoted segmental (Kingsbury) type. This was the first use of this type of thrust bearing in the navy, but its use has since become practically universal. The single helical gear has not been used in the later destroyers, which are of much greater horsepower (requiring longer pinions), but it has proved very satisfactory on the *Wadsworth*.

The *Wadsworth* really marked the end of the era of direct-connected turbines, although several destroyers were built after this date with direct-connected turbines. During the war it was not possible to obtain gears in such great numbers as they were needed, so a certain number of direct-connected turbines were installed. These reached a high degree of perfection in the boats turned out by Newport News Shipbuilding & Drydock Company, as these boats made over 35 knots on their full power trials.

BOILER CAPACITY INCREASED TO OBTAIN HIGHER SPEEDS.

The next step in the evolution of the destroyer was to increase the speed by a large increase in boiler capacity. These boats were built during the war under great pressure and various expedients were considered in the hope of expediting production. It was even considered reducing the

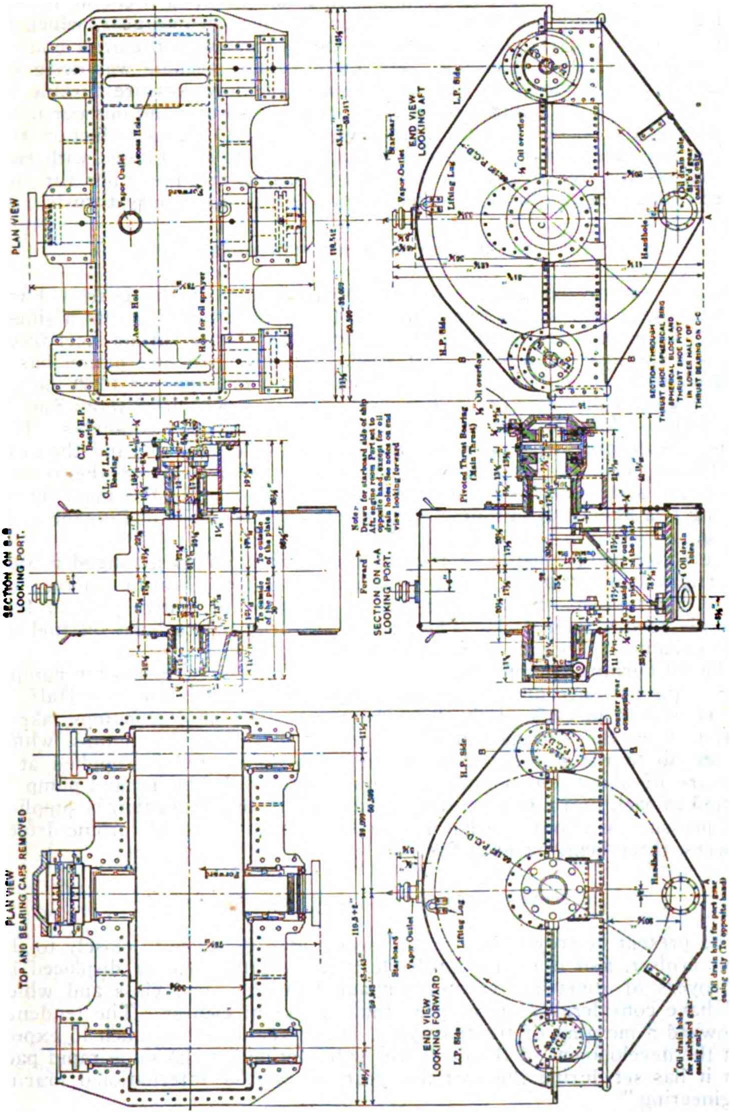


Fig. 20.—Details of Reduction Gears

speed, reducing the number of boilers, etc., but none of these offered any hope of greatly increased production, so it was decided to build the best boat possible under the conditions. The data on this class of vessel are given in the first page of this article in comparison with the *Cushing*.

The propelling machinery is located in two separate compartments, one forward of the other. It consists of geared turbines of the Parsons type, there being a high-pressure and low-pressure turbine in each engine room; the high- and low-pressure turbine pinions mesh with one gear, which is on the propeller shaft. The backing turbine is located in the after end of the low-pressure turbine casing. The high-pressure turbine was designed to run at 2,988 revolutions per minute, and the low-pressure turbine at 1,775 revolutions per minute, for a shaft speed of 436 revolutions per minute, giving a reduction for the high pressure of 6.85 and a reduction for the low pressure of 4.07. The high-pressure turbine is arranged with two connections for live steam, one being for cruising and the other for full power. The turbine thrust is taken by two 9-inch Kingsbury thrusts.

CONSTRUCTION OF REDUCTION GEAR.

Fig. 20 shows the general arrangement of one of the gears. Each reduction gear consists of two double helical pinions meshing into a single gear. The pinions are of nickel steel; the gear rims are carbon steel forgings and the gear centers are cast steel. The high-pressure pinion has a pitch diameter of 9.91 inches, the low pressure a diameter of 16.69 inches, and the gear a diameter of 67.96 inches. The total length of tooth face is 37⅞ inches. The spiral angle of the teeth is 39 degrees 53 minutes. The high-pressure pinion has 41 teeth, the low pressure 69 teeth, and the gear 281 teeth. The oil for lubricating the gears is taken from the forced lubrication system and is supplied through spray nozzles. The main thrust bearing is a 21-inch Kingsbury bearing with its housing built into the gear casing.

The boiler plant consists of four White-Forster boilers arranged in two watertight compartments. The boilers have a total heating surface of 27,000 square feet. Each boiler has an independent smoke pipe 20 feet high above the main deck. The boilers are designed to burn oil fuel on the mechanical pressure atomization principle.

The oil-burning equipment consists of two light-service booster pumps, four heavy-pressure duplex service pumps, and two oil heaters. Half of this is located in each of the two firerooms. The booster pumps take a suction from the storage tanks and deliver to the service pumps, which deliver oil to the burners through the oil heaters. Oil is supplied at a pressure of about 250 pounds per square inch. A small hand-pump is located in each fireroom for raising steam. Air for combustion is supplied at a pressure of about 8-inch water pressure by six vertical turbine-driven blowers, three being in each fireroom.

DESTROYER APPROACHING SCOUT CRUISER TYPE.

The present destroyer, in size and speed, approaches very closely to the scout cruiser, and it is very probable that the latter will be displaced by destroyers of somewhat greater tonnage than the *Dent* class and which will have considerably more speed than the scout cruiser. The tendency is toward a merging of the two types. However, it is too much to expect that the development of the destroyer will continue at the same rapid pace that it has set during the past five years of war.—"International Marine Engineering."

TRAINING OF ENLISTED PERSONNEL OF THE UNITED STATES NAVY FOR ENGINEERING DUTIES.

By Lieutenant Commander R. R. Smith, U. S. N.

At the outbreak of the war the Bureau of Navigation of the Navy Department was confronted with the problem of supplying trained engineer personnel to man not only the vessels of the Navy in reserve, which were then being rushed into active commission, but the interned German and Austrian ships which were completing the repairs necessitated by the sabotage they suffered during their internment. New construction in excess of anything heretofore attempted was being planned, while construction under way was being rushed to completion. Deck personnel was required also, but, as Navy training has always chiefly concerned itself with the training of its deck personnel, facilities and methods were already available for this training and only required expansion. The training of the engineering personnel has, however, always been a slower process on account of the greater degree of skill required in the handling of mechanisms. Our merchant marine was small and the comparatively few merchant engineers could not be spared to the Navy. Naval recruits by the thousands could be sent to these ships, but the sending of green men to do the work of trained engineers and firemen was rejected as impracticable.

A conference was called at the Bureau of Navigation, which was attended by representatives of the commander-in-chief of the Atlantic fleet. The representatives from the fleet proposed a scheme which meant a new departure for the Navy. It required setting aside a great many old ideas, but it sounded good. The conference adjourned until the next day, when the details which had been worked out over night were presented. They were accepted, and at the suggestion of the chief of bureau it was decided that an officer should be ordered to the staff of the commander-in-chief to follow up the development of the scheme proposed.

THE PROPOSED SCHEME.

The salient features of the proposed scheme as outlined in the commander-in-chief's letter were as follows:

(1) Eight battleships of the older type to be assigned to the paramount duty of training engineers.

(2) Reduction of deck complement to one-half to make room for the increased number of engineers.

(3) Greatly increasing the engineer petty officers to provide suitable instructors.

(4) Removal of practically all firemen from ships assigned to this duty.

(5) Increasing the number of trained engineer officers for each engineering ship.

The only two ships available at the time were the *Alabama* and the *Kearsarge*. Later six more ships joined the training squadron. These ships were ordered to revise their complements immediately and to transfer all engineer personnel except ninety men. Additional petty officers to make up the ninety in the required ratings were transferred from battleships not engaged in this work. All battleships were to regard one-third of their engineer complement as available for transfer, being obliged to train recruits to this number for their own vacancies.

A senior assistant to the engineer officer and two commissioned officers of engineering experience for training officers were ordered to each training ship. The complement of petty officers was as follows:

Chief machinist mates	9
Chief water tenders	9
Machinist mates, first class	9
Water tenders	27
Oilers	18
Firemen, first class	18
Total	90

The firemen were for the ship's use on evaporators, ice machines, steamers and other auxiliaries where some permanence was essential. Each ship was required to train the following men at one time:

Engine drivers	16
Oilers	32
Water tenders	32
Firemen	200
Total	280

There was thus a total of 370 engineers on board, not including the electrical force, which brought the total to nearly 400. The regular engineer complement of the *Alabama* for war conditions was 170, and during peace much less than this. In order to facilitate the work of receiving, stationing and transferring so large a number of men, it was decided that half be transferred at a time. Six weeks was decided upon as the training period. In order to alternate the transfer and commence the system, half of the men had to be transferred at the end of the first three weeks. The better half were selected, this being possible owing to the fact that the first men received came from the other ships, had been on board anywhere from one to three months and had got some training before being received on board.

Upon the internal organization of the ship depended the success of the entire scheme. It was realized that the moment the individual man under training did not receive close personal supervision by some one person, the entire scheme would fall down. The entire organization was based on this idea and the squad system was inaugurated.

It has been stated that no one person can intelligently supervise the entire work of more than eight men; when the number is increased beyond this the closeness of the supervision is relaxed.

SQUAD SYSTEM INAUGURATED.

The men under training were divided into four companies. Each officer had two companies, each company was commanded by a chief petty officer. The company consisted of seventy men under training, divided into ten squads, each squad being commanded by a petty officer not below the rating of first class. Two of these squads were engine room squads, of six men each, with a machinist mate in charge of each. One squad was in training for water tenders and was in charge of a chief water tender; seven squads were in training for firemen and each squad was in charge of a water tender.

Diagrammatically expressed, the organization was made up as follows:

S. E. O.

1st Training Officer — 2d Training Officer

1st Co. CPO — 2d Co. CPO — 3d Co. CPO — 4th Co. CPO

CMM	MM1c	CWT	WT	WT	WT	WT	WT	WT	WT
6	6	8	8	7	7	7	7	7	7

Each engine room squad of six men was made up of two men in training for engine drivers and four men in training for oilers. The ship's oilers were utilized as instructors on watch, but not as squad leaders. In conducting the training the squad system was adhered to throughout.

DEPOT DIVISIONS ESTABLISHED.

When the scheme was first put into operation each ship was assigned a Depot Division. Each battleship division in squadrons three and four were assigned two training ships for the purpose of transfer and receipt of men. The trained men were sent back to the Depot Division ships, which in turn transferred them where needed. A steady inflow of recruits was supplied to the Depot Ships, to supply the needs of the training ships.

Recruits without previous experience were sent for training as firemen, firemen of naval experience were sent to be trained as water tenders, while recruits having mechanical experience in civil life were sent for training as oilers and engine drivers. The latter term was devised to supply men who could operate machinery. It was thought that men could be made available for the specific duty of handling an engine much more quickly, if their training was confined to this particular subject. It was realized that any attempt to train men for general duties in the engine room would require a great deal more time than was available. The need for men was urgent and pressing.

RECRUITS OF MARKED INTELLIGENCE.

Much of the success of the training was directly attributed to the fact that the recruits were men of an unusually high order of intelligence, nearly all of them having enlisted for voluntary service upon the outbreak of war. Many excellent mechanics were enlisted under the rating of machinist mate, second class, and, while they had no marine experience, they made excellent progress. Among the firemen were found many men who could have been enlisted as machinist's mates. Most of them, however, had rushed into the service in any rating. No difficulty was experienced in obtaining men for the engine room branches, but it was soon found that the supply of firemen of experience for training as water tenders was running low. A water tender can learn his regular duties without much difficulty, but experience in fireroom mishaps is essential as an element of safety. In training new men for this rating it was decided that they would not be rated, but a notation made in their records that they had been trained for this work.

The men showed the greatest enthusiasm for their work. They realized that everything possible was being done to instruct them in their future duties; that they would soon be cast upon their own responsibilities and would be expected to produce results. Men who received a certain mark were recommended for one higher rating; in some cases men of unusual ability were jumped from third class firemen to oiler, machinist mate second class or firemen first class. After a new draft had been on board one week and some knowledge of the personal qualifications of the individuals had been obtained, it was necessary to shift a few men about so as to give them an opportunity to qualify for the more expert ratings, while some men in training for the higher rating were shifted to the firemen squads.

REPAIR WORK DURING NIGHT WATCHES.

During the first months of training the ships got under way at dawn and steamed until sunset, banking fires over night. On the *Alabama* all repair work was undertaken during the night watches. Feed pumps were opened, condenser tubes repacked, bearings adjusted, and everything closed up before four A. M., when steam was turned on the main steam line for warming up. We welcomed the opportunity to repair something and required the men under training to do all the work under the supervision of the petty officers. It is surprising how quickly a man can learn to do something if he is allowed to do it. While under way all the firing was done by the men under training under the supervision of the squad leader. The water was tended by the men under instruction for that rating, under the supervision of the squad leader, a chief water tender. The engines were handled and oiled by new men. It was difficult to persuade the old men that they must allow the recruits to do everything, but they finally got the idea, and later rather enjoyed it.

On a Monday morning after the receipt of a new draft the ship would be got under way by an entire crew of green men. Sometimes, upon getting under way, a signal of one-third speed ahead, followed by another of two-thirds ahead, and another of standard speed would be received from the bridge before the student could get the engine turned over. But no one was allowed to interfere, and unless the situation on the bridge was known to be urgent they were allowed to wait and suffer the delay. It must be said that the commanding officers made the best of the situation, and, while regretting the subordination of the military mission of their ships, lent their heartiest co-operation to the engineer officers.

As the interned ships were completing their repairs, the Depot Ships were discontinued and the men sent directly to the receiving ships. Each draft was detailed specifically for the ship to which it was going and the interest ran high. It was necessary in self-defense to obtain and post all data available on the ships in question to satisfy the curiosity of the men. By so doing it lent interest to the training. When the *Vaterland*, now the *Leviathan*, was commissioned, each man wanted to be sent to her. When there was no objection, each of the top men in their squads were allowed to ask for the ship they wanted, and their request was granted, if possible. The men in the deck force commenced going to the mast, requesting to be put in the training draft. Some of these requests were granted.

MEN SENT TO CAMPS AT NAVY YARDS.

After the commissioning of the interned ships, the men who completed the training course were sent to the camps at the various navy yards. The

new drafts were received directly from the receiving ships and this method was steadily followed.

As stated above, the training ships operated during the day only during the first months of training. Later, however, all ships operated with the fleet, and the training afforded the men below, while not so good for the engine drivers, was better for the firemen, oilers and water tenders, inasmuch as the ship was operating under conditions more closely approximating the conditions the men were to meet. It gave the firemen experience in cleaning fires, which was not necessary when fires were banked each night. At the same time the upkeep diminished rather than increased, as might have been expected from the nature of the work upon which the ship was engaged.

The watch standing was supplemented by instruction periods, twice a day for the men not on watch. The schedule was so arranged with regard to the watches that all men in training for the same rate received the same instruction.

The following courses of instruction were given. These were supplemented by lectures on naval usages and customs of the service:

COURSE OF INSTRUCTION.

ENGINE DRIVERS AND OILERS.

Feed Water Supply.

(1) Height of water—methods of ascertaining.
(2) Supply of feed water—how obtained.
(3) Action necessary at failure of feed water.

Main Engines.

(1) Throttle valve, butterfly valve, by-passes, bleeders.
(2) Reversing engines, link gear and cut-offs.
(3) Cylinder drains and relief valves.
(4) Gage board.
(5) Bearing, main, thrust, and steady. Sterntube.
(6) Piston rods, valve stems, crossheads, gibs and guides.
(7) Engine and fireroom telegraphs and counters.
(8) Vacuum and loss of same.
(9) Oil—grades used for different purposes.
(10) Oil service and water service.

Main Engines and Auxiliaries.

(1) Feed pumps, main and auxiliary.
(2) Air pumps, circulating pumps and condensers.
(3) Feed heaters, separators and traps.

Feed and Steam Lines.

(1) Feed tanks, reserve tanks, pumps, connections, feed lines, main and auxiliary.

Ship's Drainage, Fire Main and Flushing System .

WATER TENDERS AND FIREMEN.

Boilers and Firerooms.

(1) Coal tally and passing coal.
(2) Priming, lighting off, steaming level.
(3) Firing.
(4) Cleaning fires and ejecting ashes.
(5) Care of bilges and strainers.
(6) Feed water—height of, supply, how obtained, action necessary for failure of water.

(7) Bunker gases—safety precautions.
(8) Time firing device.

Feed Lines.

(1) Feed pumps, main and auxiliary.
(2) Feed heaters, separators and traps.
(3) Feed tanks, reserve tanks, pumps and connections.

Ship's Drainage and Flushing System—All Boiler Connections and Fittings.

(1) Safety valves, feed checks and stops, bulkhead stops.
(2) Steam traps and cut-outs.
(3) Water columns and try cocks.
(4) Drums, handholes, manholes, door casings.

Each subject was developed by a questionnaire for the use of the squad leaders, and there is in use on board at the present time three pamphlets, one for each course. An example of one of these lessons is given below:

COURSE FOR ENGINE DRIVERS AND OILERS.

Lesson No. 12.

Subject.	*Reference.*
1. Feed Water System.	Ship's Installation. Eng. Handy Book, pp. 85, 155, 158 and 161.

Questions may be made up for this lesson from references given.

Q. —Describe the following parts of the feed water system, including the material of which they are made.

How do they operate?
How are they connected to other parts?
How are they cleaned?
What care and preservation is required?
What troubles are encountered?

1. Reserve feed tanks.
2. Feed tanks.
3. Feed tank filters.
4. Feed pumps.
5. Feed heaters.
6. Grease extractors.[1]
7. Feed lines, main and auxiliary; expansion bends.

Q. What is back pressure? Where is it used? How much is carried? How is it regulated? What is a hot well? Where is it? Why? What is in it? Why? How often are filters cleaned? Trace the main feed line from the feed tank to the boilers, naming all the valves, pumps, expansion bends, connections and cross-connections and other auxiliaries in the line. Why are there two feed systems? Tell where they are and what cross-connections are between these two lines. When would you cross-connect them? How would you put feed water into the reserve tanks? How would you get it out?

The course was developed on the *Alabama* into the series of lessons of which the above is an example. They were made as simple as possible in order that all might understand them. The course as published is in use on all the training ships. Each ship has developed the subjects, while several have adopted the *Alabama* pamphlets.

At the conclusion of the course a page was inserted in the enlisted record of each man, giving the following information:

INTENSIVE TRAINING—ENGINEERS.

(C-in-C Letter No. 1885 of 4-24-17.)

Trained on board U. S. S. ....
Name ....
Actual rate ....
On transfer from U. S. S. ....

Instructed as ....
General proficiency ....
Qualified as ....
Type of boiler...
Type of engine ....
Lubrication of engines..
Fuel ....
Received from U. S. S. ....
Date received ....
Transferred to ....
Date transferred ....

The training has been in operation for about fifteen months at the time of writing this article. Approximately 16,000 men were trained in this time. These men are now doing duty on all types of ships, including battleships, cruisers, transports, destroyers, yachts and submarines. On a recent trip abroad the writer had an opportunity of questioning a great many of the men who had passed through the training course. Many of them had been advanced in rating. This is the best indication that their training had been satisfactory. The reports received from the officers who were working with these men were very satisfactory.

What was a great problem at the beginning of the war has been solved in a very simple manner. If it had not been satisfactorily solved, it is believed that the Navy would have been seriously hampered in its operations. The absence of fireroom and engine room casualties must indicate that these men who have come into the Navy since the commencement of the war have been making good. The officers who have had a part in this work can feel that they have contributed directly and effectively towards the excellent showing of the Navy during the past year.—"International Marine Engineering."

THE PROPULSION OF SINGLE-SCREW SHIPS.

The determination of the horsepower necessary to drive any particular ship at a given speed may be made by the choice of a suitable Admiralty constant or by model experiments. In the former all the factors which comprise the performance of the ship are included, and no special calculations are required to ascertain the resistance of the ship or the efficiency of the screw. Certain advantages therefore are to be gained from using the constant methods, but these are nullified to some extent by the uncertainty as to the constant to choose. Care must be taken to fix one which from previous actual experience has been obtained in a vessel very close to the type being powered. When the model experiment method is adopted the problem is only partly solved, as the effective horsepower thus obtained forms, roughly only one-half of the total power to be installed. The choice of the proper propulsive efficiency thus becomes an important item, and this can only be done with satisfaction when the effective and indicated or shaft horsepowers of a previous vessel of similar type are known.

In the analysis of the performances of ships it has been a matter of some surprise to find that in spite of the high hull efficiency single-screw propulsion does not appear to be much superior to that with twin, triple or quadruple screws. When Sir William White many years ago gave a paper to the Institution of Naval Architects on "The Speed Trials of Recent Ships of War," this feature was referred to. Many times since then has the fact that single-screw ships do not surpass twin-screw vessels in gross propulsive efficiency been alluded to. In the case of the *Vespasian,* for example, the results of which were given by Sir Charles Parsons to the Institution of Naval Architects in 1910, the propulsive efficiency was only 53 per cent with reciprocating machinery, a performance which has often been excelled by twin-screw vessels.

It has been suggested many times that the efficiency of a good propeller should be in the region of 70 per cent. If this figure were accepted, the difficulty of accounting for the moderate propulsive efficiencies of single-screw ships becomes the more acute. Now it is probable that the majority of screws in common use may have such an efficiency when revolving at some particular slip, but it is certain that, when revolving, such screws are exerting the necessary thrust to drive the vessels to which they are attached at their particular speed, the real slip at which this thrust is given is greatly beyond the point of maximum efficiency. This point has been admirably illustrated by the latest experiments of McEntee in the Washington tank.

A certain propeller was attached to four separate models and propelled in the Washington basin. It was found, by independent experiment, that this particular screw had a maximum efficiency of 71 per cent at about 13 per cent real slip. The efficiency curve then fell rapidly, until at 53 per cent real slip the efficiency was only 50 per cent. The models were of a full cargo ship and the slips at which the screw was working when the necessary thrust was given varied from 40 to 50 per cent in the four models tried. The actual efficiency of the propeller within these limits was 60 per cent. On a basis of shaft horsepower the hull efficiency value was 1.05, the wake being greater than the thrust deduction loss. If it be assumed that the mechanical efficiency of reciprocating machinery is 90 per cent, this 63 per cent on the shaft horsepower basis would be reduced to about 56 per cent on a basis of indicated horsepower—a not unreasonable figure.

It must be remembered that, although in single-screw vessels the propeller is placed most advantageously from the wake standpoint, a large thrust deduction factor is generally associated with these high wakes.

In the measurement of wake values in a model basin it is possible that some errors arise, not necessarily in the determination of these factors, but in their direct application to full-sized ships. In model experiments the frictional resistance is a greater proportion of the total than is the frictional resistance of a full-sized ship; that is, a correction must be applied to the model resistance in order to arrive at the proper ship resistance. If, then, the frictional resistance is magnified in model results, is it not more than likely that the frictional wake is also greater than would obtain in the actual vessel? What the difference would be is problematical, but it undoubtedly would have the effect of reducing the hull efficiency as measured by tank methods and go in the direction of accounting for the obtained propulsive efficiencies in single-screw ships.

The main point, however, is in the loss in efficiency due to the great slip at which the majority of single screws operate. When limited diameters obtain, the thrust necessary to propel a full slow form can only be produced when the screw is revolving with great slip, and this means on the descending point of the efficiency curve.

It will be interesting to learn from the results of the trials of the numerous cargo ships being fitted with geared turbines if the hopes of obtaining relatively high propulsive efficiencies with these are obtained. Unfortunately, the war retarded the adoption of this type of machinery in cargo ships, and insufficient data are at hand regarding the sea performances of geared turbine vessels. Should the hopes be realized, a great step will have been made towards the attainment of higher efficiencies and great economies.—" International Marine Engineering."

INTERNAL COMBUSTION ENGINED WARSHIPS.

Lord Fisher's recent communications to the "Times," urging economy and suggesting characteristically sweeping means to that end, have raised afresh some controversial points. The nation is not unmindful of the contribution in the past of Lord Fisher in adopting strong measures to ensure at all costs that the efficiency of the Fleet should be maintained at the highest possible value. A reminder is hardly needed of the part played in securing such efficiency of fighting force by the water-tube boiler, the steam turbine, the big gun, and the submarine, or of our indebtedness to Lord Fisher in his advocacy of these important factors. As a result, such improvements were effected in naval engineering that when the war commenced we led the world by a good margin. Lord Fisher also realized at an early date the potentialities of the internal-combustion engine as a means of propulsion for fighting ships.

This subject was first raised in a paper before the Institution of Naval Architects given by Sir James McKechnie on March 20, 1907. Later on, it will be remembered, a Royal Commission was set up at the instigation of Lord Fisher, and presided over by him. The findings of this Commission have not been made public, but it was generally recognized that the report was a disappointment to Lord Fisher to the extent that it indicated that at the time when the inquiry was held a great amount of development still remained to take place before the internal-combustion engine could be regarded as assailing the first place in naval propulsion firmly and justly occupied by the steam turbine. The Commission certainly served a useful purpose by the collection of evidence and data which showed clearly the state of the internal-combustion engine at that time and indicated the work to be undertaken and the problems to be solved before a reconsideration of the application of the Diesel engine to naval work of high power need arise. Oil having been proved in practice in the interval to be the fuel *par excellence* for naval use, and the Diesel being that type of internal-combustion engine best suited to utilize that fuel with maximum economy at relatively high powers (for this class of prime mover) our present remarks may be confined solely thereto.

Lord Fisher's recent letters to the press have given rise in the minds of many engineers to that reconsideration. During the war experiments on such subjects would not have been justified, and reliance was placed rather on combinations of well-tried units, changes being made only after full consideration of the maximum possible detrimental effect on fighting efficiency should the anticipated improvement be completely negatived. Such, however, was the mass of the production of naval machinery undertaken during the war that short steps carried us relatively long distances, and in effect the progress made during this period in most of the branches of naval engineering was greater than in any previous period of time of equal length. The one exception is undoubtedly the Diesel engine. The same 100 brake horsepower per cylinder that was standard in our sub-

marines before the war was the standard, except for minor improvements, at the end. The Germans in their submarines had engines of some 300 brake horsepower per cylinder (see "Engineering," "German Submarine Diesel Engines," June 13 last, page 763), and the Clyde during the war produced a merchantman with two main engines of 16 Diesel cylinders, developing in normal running at sea a total of 6,600 indicated horsepower, or over 330 brake horsepower per cylinder. The engines referred to are the four-cycle single-acting Diesel engines of the motor ship *Glenapp*. The cylinders have a diameter of 760 mm. (29.92 inches) with a stroke of 1,100 mm. (43.31 inches) and run at 125 revolutions per minute. These are not experimental engines, although much work of such a nature has been undertaken and remains to be recorded. Before the war many rumors concerning Diesel engines building on the Continent for battleships were current. The facts are briefly as follow: The M.A.N. Company, at Nuremberg, built a three-cylinder two-cycle double-acting engine, from which several thousands of horsepower were expected. The results, however, were disappointing, and a serious accident—an explosion in the scavenging system setting fire to the staging and screens erected round the engine for secrecy—which, however, concerned only very indirectly the main principles of the Diesel engine, had the direct effect on the Continent of stopping the great amount of experimental work of very similar nature in process at that time. Messrs. Krupp in Germany built a 2,000 brake horsepower double-acting two-cycle cylinder, with which no substantial success was achieved. In France Messrs. Schneider, of Le Creusot, under license from Messrs. Carels, of Ghent, constructed a 1,000 brake horsepower single-acting two-cycle engine, regarding which no detailed results are available.

Perhaps the most successful of such units was that of Messrs. Sulzer, of Winterthur, where a single-acting two-cycle cylinder of 39.4 inches diameter by 43.3 inches stroke developed at 150 revolutions per minute, it is claimed, 2,000 brake horsepower, with a fair measure of success, sufficient, at any rate, to justify the builders in standardizing a two-cycle cylinder of 600 brake horsepower with a cylinder diameter of 29.5 inches and a stroke of 39.4 inches, running at 130 revolutions per minute. A six-cylinder engine of these units gives 3,600 brake horsepower, and of these several have been built for land work.

These pre-war examples indicate the progress made up to that date, and it must be admitted that the large experimental engines were generally unsuccessful, even although they were designed and built with the one and only aim of giving the maximum horsepower regardless of any other considerations such as weight, compactness, piston speed or speed of revolution. The explanation is to be found in that they were too far ahead of the knowledge at that time of this type of engine. In some cases the experiment may have been justified on the score that the information gained was applicable to and effected improvement with engines of moderate power. The *raison d'etre* of this pre-war campaign was that Germany, in many ways the home of the Diesel engine, had lagged behind with the steam turbine, but hoped by intensive development, government fostered, to regain, with this prime mover, the place she had lost on account of our progress meantime with the steam turbine. In this Germany failed, although her engine builders can claim that their largest war submarine engines showed latterly a power capacity per cylinder of nearly three times that achieved by the British standard engine.

Another landmark is the 400-500 brake horsepower submarine unit cylinder of 350 revolutions per minute, designed and built by Messrs. Sulzer during the war, with which highly satisfactory results have, it is

believed, been obtained. Six-cylinder engines composed of these units have been built developing 3,000 brake horsepower per engine.

These examples indicate in a general way the various steps in the development of the Diesel engine up to the present time. The question then is, can we assess a value to this progress in comparison with the total distance required to be travelled before such prime movers can be considered for first-line fighting units of the fleet? In view of the success of the steam turbine such a consideration will be regarded by some as unnecessary.

The Diesel engine possesses potentialities for fighting ships that demand repeated attention in order that they may be equated to naval requirements if and when the known deficiencies are remedied. The attractions of the Diesel engine are: Its oil economy, using the most suitable naval fuel, far surpassing the best results achieved by or predicted for the steam turbine; its readiness for immediate service without requiring warming through, etc., the relative ease with which exhaust gases can be discharged compared with these for steam plant, the better arrangements of guns and machinery due to the elimination of the uptakes and the substitution of a relatively small exhaust pipe for funnels. Against these advantages must be set against the disadvantages of small power per unit cylinder, the high weight, height and uncertainty of operation, especially with large units, under certain practical conditions.

Instancing three standard powers of naval vessels, the small battle cruiser and the battleship, and light cruiser and torpedo-boat destroyer, the first of 80,000 shaft horsepower, and the others of from 40,000 to 27,000 shaft horsepower, and assuming cylinders of 1,000 brake horsepower, 80, 40 and 27 would be required. The maximum number of Diesel cylinders in any vessel afloat is found in the "J" class submarine boats, with their 36 cylinders (three 12-cylinder engines). The maximum number of units which can be regarded as suitable is therefore relatively large. Twelve turbines and more than twelve boilers are not regarded as excessive in steam practice, and 36 cylinders in the case of internal combustion engines have, as already mentioned, proved a sound arrangement.

In respect of increase of power per unit with Diesel engines, progress, it will be seen, has been relatively slow. Up to 300-400 brake horsepower can be said to be proved at the present time to be practical politics. Higher powers still can no doubt be made to give quite satisfactory service, but the demand hitherto has been severely limited by the remarkably rapid success of the application of geared turbine machinery for vessels of all powers, down even to relatively small installations. For naval work, however, the advantages of internal combustion create an insistent demand.

Success in the past with increasing power of Diesel engines has not been achieved without encountering many difficulties. Chiefly, it must be stated, on account of insufficient attention being paid to small and intricate but absolutely essential details, of which with this type of engine there are so many. Knowledge is increasing, and a satisfactory cylinder developing 1,000 brake horsepower may confidently be anticipated in the near future. Whether, however, it will meet naval requirements or not is another matter, depending essentially on considerations of weight, space occupied, and such factors. We discussed the future of this type of prime mover in regard to these terms in an article entitled "Internal-Combustion Engines for Submarines and Aircraft,"—published in "Engineering," of September 20, 1918, page 319—and held that the future would see in such work much progress, as has been the case with other prime movers, notably the petrol aero engine. The internal combustion turbine also has been discussed, but still remains no more than a fervent hope.

It is too early yet to speak in any definite way of future naval policy, yet, in all probability, the efforts of the next few years, at any rate, will be concentrated upon the development of those types of fighting units that have proved themselves or have indicated the path of progress. This policy will entail considerable experiment, in which the Diesel engine will not be overlooked. Such work is always costly, and whether or not the results are dearly bought depends largely upon the method of control, the imagination brought to bear, and the careful consolidation of each step gained.—" Engineering."

DEVELOPMENT OF GEARED TURBINES FOR THE PROPULSION OF SHIPS.

By Mr. R. J. Walker.

Although the successful application of the steam turbine to marine propulsion dates back to the year 1897, when Sir Charles Parsons demonstrated in that now historical vessel, the *Turbinia,* the great advantages of the turbine system when applied to the propulsion of ships, it is only within the last few years that mechanical reduction gearing has been largely adopted in association with steam turbines.

Sir Charles Parsons always had in mind the possibility of reduction gearing in connection with the application of his steam turbine to marine propulsion, and even as far back as 1897 a small launch was constructed in which helical gearing similar to that adopted by Dr. De Laval, of Stockholm, with his high-speed turbines for land purposes, was introduced between the turbine and the propeller. This is believed to be the first application of helical gearing to drive a propeller.

It was considered desirable, in view of the very wide field which was opened up by the success of the *Turbinia* for the application of direct-driven turbines to high-speed vessels, to confine attention for a time to development in this direction only, and to defer embarking on the additional pioneer experimental work incidental to reduction gears of unprecedented size. The wisdom of such a policy was subsequently shown by the great difficulty experienced at the commencement in introducing the turbine system of propulsion against the conservatism of British engineers and shipowners who looked upon such a revolution in the methods of marine propulsion with a great amount of doubt and distrust. Very little experience existed at that time of the accurate cutting of toothed gearing in large sizes, and the additional work involved would certainly have added to these difficulties.

Before proceeding to consider the development of geared turbines, it may be of interest to briefly review the progress made in connection with direct-driven turbines.

Following on the success of the *Turbinia,* the torpedo-boat destroyers *Viper* and *Cobra* were built and fitted with turbines for the Admiralty in 1899, and achieved remarkable speeds.

The next step in the case of war vessels was when the system was fitted to the third-class cruiser *Amethyst,* three other sister vessels being fitted at the same time with reciprocating engines. The results of these trials demonstrated the economy of the turbine in this class of vessel, more especially at the higher powers, and influenced the British Admiralty in deciding to have the battleship *Dreadnought* fitted with turbine engines. The keel of this vessel was laid down at Portsmouth in October, 1905. By 1906 the British Admiralty had adopted turbines for all new construction.

In the early endeavors to introduce the turbine for commercial purposes.

no shipowner could be found bold enough to make the experiment on any reasonable terms. It was ultimately recognized that the only chance of convincing shipowners of the practicability of such a proposal was to build a vessel specially for the purpose, and Captain John Williamson, Messrs. Denny Brothers, and the Parsons Marine Steam Turbine Company, Limited, undertook the financial responsibility of building the Clyde steamer *King Edward,* which was first put into service in the summer of 1901, and has been running satisfactorily ever since that date. Other vessels very quickly followed the *King Edward,* until at the present time the total shaft horsepower of turbine vessels on service for commercial purposes is about 1,600,000.

One of the chief difficulties which had to be met in applying the steam turbine to the propulsion of ships arose with the propellers. As is now generally known, it is desirable that a turbine for maximum efficiency should run at a high rate of revolution, whilst on the other hand, for maximum propeller efficiency, much lower rates of revolution are necessary. Obviously one solution was by the introduction of some form of gearing between the turbine and the propeller; the time, however, for such a departure, for reasons already mentioned, had not then arrived. The problem was therefore to reconcile as far as possible those two opposing factors, and to arrive at the best compromise to meet the conditions required by a suitable lowering of the revolutions as then designed for land turbines, and a raising of the revolutions of the propeller by suitable modifications in the form. This was one of the most serious problems which had to be faced in the design and construction of the *Turbinia,* and led to many modifications in the propellers and turbines of that vessel before success was finally achieved.

With the raising of the revolutions of the propeller the diameter and pitch ratios were less, whilst the ratio of blade surface to disc area was much greater than is usual with propellers driven by ordinary reciprocating engines.

It was found as a result of numerous and costly experiments that a judicious compromise could be, and was, arrived at, whereby satisfactory combined efficiencies of turbines and propeller were obtained for high-speed vessels, resulting in increased speed of ship and increased economy in coal consumption.

The chief governing factors in marine steam turbine designs are those of economy, weight, and first cost, and it was found in actual practice that the problem of applying the turbine direct to the propeller was for the time satisfactorily solved to fulfill these conditions for vessels of about 18 knots speed and upwards.

Up to the year 1909 the steam turbine had not been applied to vessels of slow and intermediate speeds, with the exception in a few instances of the combination of reciprocating engines with a low-pressure turbine.

From the early years of steam turbine design, it became apparent that the turbine engine was capable of economically dealing with ratios of expansion beyond the limits possible with reciprocating engines, and a combination of reciprocating engines with a low-pressure turbine presented advantages as regards economy in that class of vessel where the designed full speed fell below the range suitable for an all-turbine arrangement, that is to say, for vessels of about 18 knots speed and under, the reciprocating engines in such an arrangement working in the region of pressure drop where the conditions are best suited to it, and the turbine utilizing that portion of the expansion diagram which the reciprocating engine is not able to utilize efficiently. The service results of vessels fitted with this combination system have shown that this arrangement of machinery

effected a saving of from 12 per cent to 14 per cent in coal consumption as compared with similar vessels fitted with quadruple engines.

It was not until the success of the marine steam turbine had been assured by its rapid adoption in vessels of moderate and high speed, and more time was available, that attention was directed to the application of the turbine to slow speed vessels by the introduction of gearing between the turbine and propeller. Several forms of gearing have been proposed, such as electrical, hydraulic, and mechanical. Electrical and hydraulic transmission gear have been fitted in a few ships, but the greater majority of vessels have been fitted with mechanical gearing.

In view of the success obtained by Dr. De Laval, of Stockholm, with helical and double helical gear in connection with his turbine for land purposes for powers up to about 600 brake horsepower, Sir Charles Parsons decided to test turbines mechanically geared to the screw shaft. Even at this stage there was encountered a very strong prejudice to gearing, and it was found necessary, in order to test its practicability for marine work, to carry out experiments in a cargo boat.

In 1901 an old steamer, the *Vespasian,* was purchased and fitted with geared turbines in place of her original triple-expansion reciprocating engines, but before the change was made exhaustive trials were carried out with the original engines. Full particulars of the results of the comparative trials of the two systems of machinery were given in a paper by Sir Charles Parsons read before the Institution of Naval Architects in 1910. It will be sufficient here to state that an additional economy in coal consumption of over 15 per cent was obtained in the *Vespasian* by the substitution of geared turbines for reciprocating engines, and that the efficiency of the mechanical gearing was found to be about 98½ per cent. The *Vespasian* was run between the Tyne and Rotterdam, carrying coals, and in general trade for four years, and the hull being then worn out was broken up, but the turbines and gearing, which showed no signs of deterioration, were taken out of the vessel and fitted to a new steamer, the *Lord Byron.*

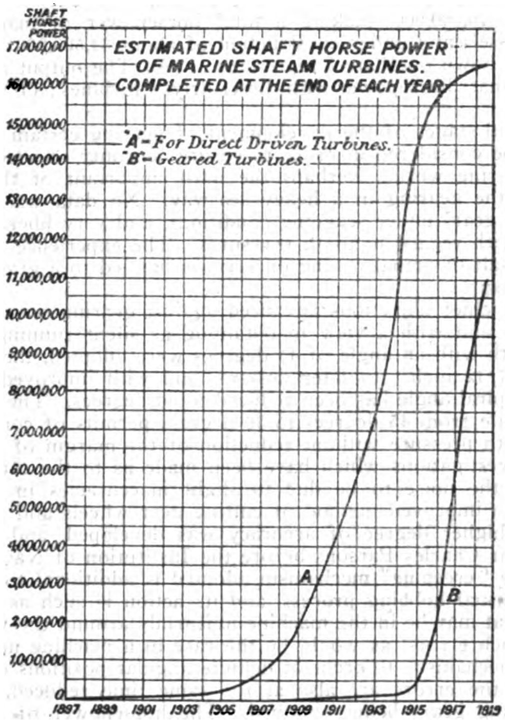
The success of the *Vespasian* aroused very keen interest amongst engineers and shipowners, and Professor Sir John Biles was amongst the first to recognize the advantages of increased efficiency to be derived from geared turbines for cross-Channel steamers when he recommended the London and Southwestern Railway Company to adopt geared turbines in their new steamers *Normannia* and *Hantonia,* for their Southampton to Havre service, and built by Messrs. the Fairfield Shipbuilding and Engineering Company, Limited. The trials of those vessels, which took place in the early part of February, 1912, were successful, and the ships have continued to give very satisfactory results.

The *Paris,* another cross-Channel steamer for the London, Brighton and South Coast Railway Company, was built a year later by Messrs. Denny and Brothers, and fitted with geared turbines, and attained a speed of 24¾ knots. After the success of these vessels, other shipowners quickly followed the lead of these companies.

Although gearing was primarily introduced to widen the field of operation by its adoption for vessels of low speed, it was soon recognized that increased efficiency could be obtained by means of reduction gearing in fast ships. The Admiralty placed an order in 1910 for the part gearing of the machinery of H.M. torpedo-boat destroyers *Badger* and *Beaver,* by introducing gearing for the high-pressure and cruising turbines of these vessels. In 1912 the Admiralty adopted gearing for the whole of the machinery of the two destroyers H.M.S. *Leonidas* and *Lucifer,* of 22,500 horsepower, fitted on two shafts.

Exhaustive trials of these vessels were carried out, the results of which showed a considerable increase in efficiency both at full and cruising speeds as compared with previous torpedo-boat destroyers with direct-driven turbines.

The adoption of all-geared turbines on two shafts in these destroyers permitted an increase in the propeller efficiency of about 12 per cent, and an additional improvement in the steam consumption of the turbines, of about 10 per cent at full power, and about 30 per cent at one-tenth of full power, also a slight saving in the total weight of the machinery, as compared with a twin screw arrangement of direct-driving turbines hitherto adopted in these vessels.



During the war gearing was universally adopted for light cruisers, battleships and battle cruisers of the highest powers, and during the last four years practically no direct-driven turbines have been fitted. Vessels are now in service with installations of geared turbines of 100,000 horsepower, and the horsepower transmitted through a single gear wheel has reached 25,000, and the power through a single pinion 15,500.

The further adoption of geared turbines in the mercantile marine was considerably retarded during the war. When the question of type of machinery came under discussion for the standard ships, preference was

given to reciprocating engines owing to the fact that all the usual builders of marine turbines were full of orders, and were required to give preference to the manufacture of turbines for war vessels, which limited the number of turbine engines which could be built for commercial vessels, and also to the fact that the country had great resources at its disposal for the quick manufacture of ordinary triple-expansion engines. In the later standard ships of the fabricated type, however, geared turbines were adopted.

At the present time the total horsepower of geared turbines for vessels for commercial purposes completed and under construction is about 1,400,000; and the total horsepower of geared turbines for war and commercial purposes completed and under construction is about 18,000,000. Previous to the adoption of geared turbines the total horsepower of direct-driven turbines was 16,500,000, making a total horsepower of marine turbines fitted into vessels and under construction of about 34,500,000.

The curve shown on Plate I represents: (A) The output at the end of each year in shaft horsepower with direct-driven turbines; (B) with geared turbines.

In the initial stages of the development of gearing certain factors were required to be considered in order to ensure certainty of success with the gears for marine work. Perhaps the most important of these was the behavior of the gearing in a heavy sea-way. No data was available as to the life of gears under seagoing conditions, and very liberal allowances were made with regard to tooth pressures. The experience gained, however, as a result of actual runing on service, has led to a gradual increase of tooth pressures.

One of the chief objections advanced against gearing was the anticipation of noise and with a view to obtaining as silent running as possible fine pitch teeth with an angle of 45 degrees were adopted, the pitch of the tooth being 7/12 inch. In later designs, and with improved methods of cutting, the spiral angle has been reduced to 30 degrees. The reduction of the spiral angle from 45 degrees to 30 degrees permits of considerable increase of tooth pressure without reduction of the margin of safety.

Careful investigations which have been made as to the causes of noise have shown the noise to be due to slight inaccuracies in the pitch of the teeth. An improved method of cutting gear wheels and pinions so as to obtain a higher degree of accuracy was developed, and described in a paper by Sir Charles Parsons before the Institution of Naval Architects in 1913. The "creeping" mechanism adopted is additional to the machine used in the usual hobbing process, and its action is such as to distribute any errors that may be in the machine uniformly around the wheel instead of leaving such errors, as would be the case in a machine not fitted with "creeping" mechanism, to occur at definite angular positions on the wheel. Incidentally, the errors are also at the same time reduced, resulting in quieter running gears being obtained. The horsepower of gears cut by machines fitted with the "creeping" tables represents about 75 per cent of the total in use.

Within the last three or four years a further development has taken place in the case of mercantile vessels by the adoption of double-reduction gearing instead of single-reduction gearing, which had previously been the practice.

Double reduction gearing permits of larger ratios between the revolutions of turbine and propellers being obtained without excessive size of gear wheel, so that even in very slow-speed vessels the turbines may be designed to run at the speed for maximum economy. The increased efficiency obtainable with a double reduction scheme and higher speed

turbines as compared with turbines and single reduction gears is in some cases as much as 7 per cent.

From first to last a considerable advance has been made in the efficiency of the steam turbine. In 1892 a turbine of 100-kilowatts gave a steam

TABLE OF COMPARATIVE RESULTS.

Parsons Single Reduction Geared Turbines and Reciprocating Engines.

NOTE.—All comparisons are for ships of same sizes and models and same boilers.

Name of Vessel.	Type of Machinery.	Dimensions and Tonnage.	Approximate Displacement.	Speed in Knots.	Coal Consumption.
			tons.		tons per day.
(1) "Vespasian"	A.—Turbines, single-screw ..	275 ft. × 38·75 ft., 2,150 gross	4,350	9½	14·17
	B.—Reciprocating, single screw	275 ft. × 38·75 ft., 2,150 gross	4,350	9½	17·07
(1) Natural draught: Result = 17 per cent. less consumption by turbines.					
(2) "Cairnross"	C.—Turbines, single-screw ..	369·75 ft. × 50·75 ft., 4,000 gross	10,000	10·15	27·916
"Cairngowan"	D.—Reciprocating, single-screw	369·75 ft. × 50·75. ft. 4,000 gross	10,000	10·15	32·692
(2) Natural draught: Result = 14·93 per cent. less consumption by turbines.					
(3) "Mahanada"	E.—Turbines, single-screw ..	7,200 gross	13,500	12·138	54·8
"Malakuta"	F.—Reciprocating, single-screw.	7,200 gross	13,500	11·538	67·44
(3) Natural draught: Result = 19 per cent. less consumption by turbines.					
(4) "Cumberland"	G.—Turbines, twin-screws .	9,500 gross ..	15,600	13·11	77·5
"Devon"	H.—Reciprocating, twin-screws	9,500 gross ..	15,600	12·8	91·5
(4) Howden's forced draught: Result = 15·3 per cent. less consumption by turbines.					

consumption of about 27 pounds per kilowatt, whilst later in larger sizes, say of 3,000 kilowatts, a steam consumption of under 15 pounds was obtained. In 1907 there was a further reduction to 13.2 pounds per kilowatt, and at the present time steam consumptions for large land installa-

tions as low as 10.3 pounds per kilowatt-hour with 275 pounds steam pressure and 200 degrees of superheat, which is equivalent to about 7.5 pounds per brake horsepower, are being obtained.

In marine work, for the earlier turbine vessels the steam consumption of the turbines worked out at about 15 pounds to 16 pounds per shaft horsepower, and in later direct-driven turbines under 12 pounds per shaft horsepower was obtained, whereas at the present day, with double reduction gearing and reaction turbines, consumptions of under 10 pounds per shaft horsepower can be obtained with saturated steam, and under 8 pounds per shaft horsepower with a superheat of 200 degrees F. From these figures it will be readily seen that a remarkable advance has been made in increasing the efficiency of the turbine, the steam consumption being about one-half of what it was eighteen or twenty years ago.

By the courtesy of several shipowners Table A has been prepared, giving particulars of the performances of vessels with Parsons geared turbines, as compared with sister ships fitted with reciprocating engines.

There are several varieties of turbines now in use, some of them only, however, to a limited extent, but within the scope of the paper it is not possible to make more than a few general remarks.

The various turbines may be classed under the headings "Impulse" and "Reaction," and in some cases a combination of the two. The reaction class has low steam velocities. In modern practice the blades of a reaction turbine are of a reasonable length and the maximum possible efficiency is higher than obtainable with any other type.

The question of maintenance of efficiency and economy in wear and tear under long continued service is a very important one to shipowners. The reaction turbine is the only system which has been tried over a long period of years in mercantile vessels. The result of such experience points to the fact that, with ordinary care, the economy of the reaction turbine is fully maintained after years of running. This maintenance and economy is due to the low steam velocities employed. It has been found that with high steam velocities there is a tendency for the blades to suffer erosion, probably caused by drops of condensed water mingled with the steam, the action increasing rapidly with increase of velocity. Blades which have been eroded show a falling-off in efficiency, but this erosive action is reduced by the adoption of superheated steam.

As regards the life of gearing, there appears as the result of experience to be no reason why it should not have a very long life, provided, of course, that ordinary precautions are taken in the manufacture and cutting of the gears, and that efficient lubrication is provided. In this connection it may be mentioned that the gearing of the *Vespasian* shows practically no signs of wear. The *Cairnross*, which was the next vessel to the *Vespasian* to be fitted with geared turbines, has unfortunately been sunk by enemy action. The *Mahanada* has now been four and three-quarter years on continual service, and from a report made quite recently the gearing is "in as good a condition as when it first left the builders' works"; from inquiries made from the owners of this vessel, the turbines and gearing during this period have not cost anything for upkeep.

On land, geared turbines have been adopted for certain work in connection with the generating of electricity. In direct-current working before the introduction of gearing, difficulties were met with owing to the fact that dynamo speeds were too low for direct coupling to turbines of maximum efficiency. Owing to the limitation of dynamo design, the turbine speed had in some cases to be reduced to less than half the speed corresponding to maximum efficiency, and in such cases mechanical gearing has been employed with considerable advantage, an economy of 15

per cent having been effected. In the case of alternators of fair size the speeds are usually such that direct coupled turbines of maximum efficiency can be used.

One of the earliest applications of gearing was that of driving a continuous running plate rolling mill for the Calderbank Steel Works of Messrs. J. Dunlop & Co. This gearing, which was of the double reduction type, and driven by a turbine of 750 horsepower, has remained in continuous operation to the present time, and a recent examination showed no signs of wear in the teeth.

Gearing has opened up a new field for the steam turbine in the driving of cotton, jute, paper and other mills.

A number of mills, both in Great Britain and in the Colonies have adopted this method of driving with most satisfactory results, both from the point of view of economy and reliability.

In conclusion, it may be safely said that the introduction of the steam turbine, has added to the progress of the world, since the turbine has considerably cheapened mechanical power on land and sea, and has rendered possible speeds greatly in excess of those obtainable with reciprocating engines. Already it has entirely superseded the reciprocating engine for war vessels and for high speed and modern mercantile vessels, and bids fair in the near future to being applied on an extensive scale for low-speed vessels, in view of the substantial economy in coal consumption obtainable thereby.—" Engineering."

SCIENCE AND ITS APPLICATION TO MARINE PROBLEMS.*

BY PROFESSOR J. C. McLENNAN, O.B.E., F.R.S., SCIENTIFIC ADVISER TO THE ADMIRALTY.

Introduction.—In the great struggle which has just been brought to its close the introduction and use of the submarine as an agent of unrestricted warfare by our former enemies, constituted a menace of paramount and vital importance to the safety and welfare of the British Commonwealth and of its allied nations.

As early as 1915, a Board of Inventions and Research was set up in the Admiralty under the presidency of Admiral of the Fleet the Lord Fisher of Kilverstone, G.C.B., O.M., G.C.V.O., LL. D., for the purpose of developing anti-submarine devices and systems capable of coping with the menace. This board consisted of the President and Professor Sir Joseph J. Thomson, O.M., Pres. R.S.; the Hon. Sir Charles Parsons, K.C.B., F.R.S.; Sir George T. Beilby, F.R.S.; and Vice-Admiral Sir Richard Peirse, K.C.B., K.B.E., M.V.O.; and it was assisted by a panel of the ablest scientists available, as well as by a secretariat and series of subcommittees of leading naval officers and scientific experts.

In the autumn of 1917, after exhaustive researches had been carried out by this board, it was finally realized by all that the submarine problem was one of the most difficult ever presented to science for solution. It became clear that it was necessary to introduce into service practically a new system of physical science and engineering, and the Admiralty, at the instance of the Right Hon. Sir Eric Geddes, G.B.E., K.C.B., M.P., First Lord, decided to broaden the field from which scientific and engineering talent could be drawn, and set up within the Admiralty a Department of Research and Experiment under the direction of Mr. Charles H. Merz. The Board of Invention and Research was incorporated in this department,

*Paper read before the North-East Coast Institution of Engineers and Shipbuilders, at Newcastle-upon-Tyne, July 10, 1919.

and Anti-Submarine Committees were constituted in such important industrial and technical centers as the Lancashire and Clyde districts.

In the naval service, too, a special Anti-Submarine Division was organized under Captain W. W. Fisher, C.B., R.N., and in the Department of Torpedoes and Mines under Rear-Admiral the Hon. E. S. Fitzherbert, C.B., and later under Captain (nor Rear-Admiral) F. L. Field, C.B., R.N., provision was made for greatly extending by research, experiment and trial the capabilities and use of improvements in mines and torpedoes as aids to naval warfare. Steps were also taken to promote the closest and most efficient co-operation between the departments of the naval service and those of the civilian scientific staff.

Extensions were also made in their organizations by our Allies, and as a result of this great effort it may be stated that by the late summer of 1918 it became clear to those associated with the movement, that the submarine problem was well in hand from a scientific point of view, that the character of the means of coping with the menace in an effective manner was clearly defined, and that the elimination of the pest by the service was only a matter of a few months' time.

At the time the Armistice was signed, only a few of the most powerful and effective devices developed were actually in service, but the results which were obtained by means of these for the short time they were in use were proportionately great. It is but right to say that the destruction of by far the great majority of the enemy submarines which were rendered innocuous was accomplished by the naval service by the use on an enormous scale of the ordinary methods available to it before the war began, or which were rendered available to the service in the early years of the war. Potentially, however, science by its advances has greatly extended the basis of service methods, and though much has yet to be done in the field of development of particular systems and devices it can be said, speaking conservatively, that the submarine in its present stage of development and in the method of its use hitherto adopted by our former enemies has been overcome as a menace to our national safety and well being.

For several reasons one in my position cannot give details of a number of the measures taken and means of destruction adopted in the anti-submarine campaign, but it is in the national interest to foster and stimulate co-operation by our people in the development of science and of its application to naval and other marine problems. In the following paper, therefore, a brief survey will be given of some of the most notable advances made in dealing with certain naval and marine problems presented to us on the naval side during the war.

ANTI-SUBMARINE MEASURES AND DEVICES.

(*a*) *Listening Devices.*—As the power of the submarine in its attack is due to its invisibility, it is clear that in countering its methods must be employed which will reveal its presence and give definite information regarding its movements. Of all the physical disturbances emitted or produced by a moving submarine the pressure waves set up in water by vibrations having their source in the vessel are the ones which are propagated to, and are detectable at, the greatest distance. Efforts were therefore directed from the first to the development of listening devices. The investigations in this field were exceedingly ramified and were pressed with great vigor.

As a result great improvements were made in hydrophones. In the development of these devices microphones and magnetophones of ex-

ceedingly high sensitivity were realized, and after enormous labor, ways and means were devised for standardizing their construction and their functioning. Hydrophones were constructed and put into service which were suitable for use in water of moderate depth, and other types were made which could be used in water of great depth. In one particular type of instrument modifications and attachments were introduced which enabled one to detect with it the direction of bearing of a source of sound with a fair degree of accuracy.

But probably the method of determining the direction of a source of sound waves in water which has proved to be the best is founded on the fact that the sound wave is in the same phase at all points of its wave-front. Thus, if there were two hydrophones, in themselves non-directional, placed in the path of the incoming sound, they can be used for finding the direction of the origin of the sound if the phase difference between the sounds received can be detected. There are two ways of doing this: the "binaural" method and the "sum and difference" method. The binaural method depends on the fact that if the sound from one receiver is conveyed to one ear and that from the second to the other ear, the impression is formed that the sound comes from a certain direction, and this direction, as interpreted by the sensations experienced, changes as the phase difference is altered. It can be brought to a certain position with respect to the listener (say, to the position directly in front) either by rotating the two receivers about an axis, or by introducing an artificial delay in some form of "compensator." This binaural method, which has great interest from a psychological and physiological point of view, has been the subject of much work on the part of both British and American scientists, and in the antisubmarine campaign it was found to be of very considerable service.

In the "sum and difference" method the impulses from the two receivers are united before reaching the ear, the combined effect observed being a maximum when there is no phase difference between the waves and a minimum when the phases are in opposition.

The French Navy has developed a hydrophone or listening device known as the Walser gear, which it has found very efficient. In this device two sets of a number of sound-receiving plates or studs are mounted on two areas which are convex to the sea, one on each side of the keel, on the underside of the hunting ship's hull. By this gear the sound impulses which they receive are not transmitted by tubes, but are given freely to the air inside the ship. These sound impulses are concentrated in a focus whose position in space depends on the direction from which the sound comes. The position of the sound focus is determined by means of a funnel or trumpet, from the contracted end of which a flexible tube leads to the listener's ear. The great advantage possessed by the Walser gear is that it enables one to distinguish by differences in the positions of the sound foci between sounds coming from a distant source and those generated in the water which have no specific directional feature characterizing them. The gear can therefore be used by chasing ships when moving with fair speeds.

With most hydrophones and many listening devices, ship's noises and water noises generally seriously interfered with their effective use, and in practice the chasing ship is compelled to stop at intervals and listen when not under way. This means that in many cases the quarry is lost.

This defect was overcome in a measure by towing a directional hydrophone encased in an artificial "fish" behind a chasing ship. By adopting stream-line formation for the towed body and suitably supporting the hydrophone, water noises are fairly well eliminated. Moreover, as the

"fish" can be towed at a considerable distance behind the chasing vessel, many of the sounds emitted by the latter do not reach it, and others which do arrive are received with weakened intensity.

It is needless to say that the development of sensitive listening devices received a great impetus by the use of thermionic amplifying valves.

Although an enormous expenditure has been made in time and effort to perfect the hydrophone and other listening devices, it is realized that such instruments possess an inherent defect. If the submarine can be made noiseless in motion, this method of countering it becomes ineffective. Even now the range of hearing is not more than 100 yards in the case of modern submarines moving at 2 knots or 3 knots.

(*b*) *Echo Methods.*—Owing to the fact that it was found possible under certain conditions to render the propulsion of submarines practically silent, it became necessary to look in other directions for fundamental methods of detecting them. A system of detection which is full of promise involves the use of a beam of sound waves sent out by a chasing ship in a manner analogous to the use of a searchlight. With such beams of sound waves it is possible to sweep the seas, and when an object of sound such as a submarine happens to come within the beam, the sound waves are reflected and echo effects are obtainable. The character of the beam is, of course, determined in large measure by the frequency of the waves constituting it. The method has been employed with great success, and promises to be a very helpful agent. It can be used by chasing ships traveling at all speeds, and when applied with certain restrictions and definite characteristics it enables one to pick up and close on a submarine situated more than a mile away. The method, it is obvious, is applicable to the locating of minefields and other obstacles to navigation as well as to submarine chasing.

(*c*) *Magnetic and Electro-Magnetic Detection.*—Magnetic detectors usually require the movable system to be poised or pivoted. They can therefore be used as yet with only a moderate degree of satisfaction in towed bodies or in vessels subjected to violent mechanical disturbances. The range at which magnetic effects can be detected is moreover comparatively short. As a result of these defects the use of magnetic detection is somewhat circumscribed. Such instruments can, however, be used under certain conditions, and in particular sea areas with great effect. In the war very considerable results were actually obtained by their use.

The range at which electro-magnetic detection can be applied is greater than is possible with magnetic detection, but the method is, however, essentially a short-range one, and in many of the forms in which it has been worked out it cannot be used with success at distances greater than about 300 yards or in depths greater than about 100 fathoms.

(*d*) *Leader Gear.*—An important application of an electro-magnetic effect which was developed during the war is found in what is known as Leader gear. This gear consists of a cable laid on the bottom of the sea along the course of a narrow tortuous channel leading into a harbor or through a minefield. If an alternating electric current be passed through such a cable it is possible by means of delicate devices installed on a ship to obtain either aural or visual indications of the presence of such a cable, and by these indications the ship can be guided in safety in fog or darkness at speeds as high as 20 knots almost with as much precision as a tram-car by a trolley wire over a railway. Experiment has shown that it is a simple matter to apply this method in water of suitable depth for distances as great as 50 miles or longer.

(*e*) *Invisible Signalling.*—Research has shown that it is possible under certain conditions to utilize polarized light or ultraviolet and infra-red radi-

ations for secret signalling. With the last-mentioned type of radiation especially valuable results are obtainable over considerable distances, even in the presence of light fogs. Where it is not advisable to use wireless communication between chasing ships, infra-red signalling is of special value.

(*f*) *Wireless Telegraphy and Telephony.*—One of the most remarkable developments which have taken place in the war is in the field of wireless telegraphy and telephony. By the use of oscillating thermionic valves especially great progress has been made. It is now possible to hold conversation with ease between a land station or a ship and an airship or seaplane over considerable distances, and by this means observers on aeroplanes or aircraft can also converse with one another. With high-power installations it has been demonstrated that wireless telephonic communication can be maintained on the sea over hundreds of miles.

On the directional side of "wireless" great advances have also been made. If an aeroplane, an airship, or surface ship should send out continuously for a short interval a series of ether waves, these waves can be picked up over long distances by devices installed in a land station, the direction of the source of these ether waves can be ascertained, and in a minute or two the land station can give the observer of the emitting source his bearing within two degrees relative to the land station. With two land stations it is possible to obtain cross bearings, and the latitude and longitude of the sending air or surface ship can be determined with a high degree of accuracy.

With directional devices installed on ships it will be possible for two ships whose positions are known to communicate its true position to a ship enveloped in a fog and situated several hundreds of miles away. It is obvious, therefore, from the few illustrations which have been given that directional wireless will find a wide field of usefulness in the future in connection with the subject of navigation.

(*g*) *Explosion Pressures.*—In the early days of the anti-submarine campaign, a method of destroying submarines whose approximate location was known, was by the employment of depth charges. To use this means it was necessary first of all to know the neighborhood in which the submarine was located, and then the chasing ship would rush to the spot and drop or throw to some distance charges of explosive which detonated when they reached a definite distance below the surface of the water. The necessity of knowing the destructive zone of any given type of depth charge soon became evident, *i.e.*, to determine the radius from the exploding charge, within which a submarine would be successfully destroyed. The same information was important in the laying out and use of mine-fields. Investigations were undertaken to determine what pressures were generated by charges of different sizes and types at various distances from the place of detonation. The nature of the pressure wave was particularly important, for upon it depends the "killing power" of the charge. The laws which govern the alteration in form and power of waves generated by these explosions had to be determined in order to employ depth charges and mines in the most effectual and economical manner. The accurate determination of the velocity of propagation of the explosive waves generated was also of importance in distributing the charges, for if waves from two different sources arrive at the object in different phases, the effective crushing power may be considerably altered. When we know what the effect of different charges at various distances is—what type of pressure wave, whether a sudden intense blow lasting a few ten-thousandths of a second, or a sustained pressure during some thousandths of a second, or a series of less intense shocks, has the greater effect in de-

stroying the submarine when under water—then we shall know how best to lay out our minefields, and what size and type of charges are the best and most economical to utilize under the various situations which may arise.

A most elaborate investigation of the characteristics of explosion pressure waves has been carried out for the Admiralty by Mr. H. W. Hilliar. By this method the pressure is allowed to act on one end of a steel piston, and measurements are made of the velocity of the piston as it passes a series of points at known distances from its starting point. From these measurements it is only a mathematical operation to extract the acceleration of the piston at different times from the moment when it began to move; in other words, the time history of the pressure acting on the piston. It is not actually feasible to measure the velocity of a single piston at different stages of its travel, but you get the same information if you let the pressure act on a series of pistons and measure their velocity after they have travelled different distances. The velocity is found from the extent of crushing of a copper plug which the piston strikes when it comes to the end of its travel.

As an illustration of results obtained by the method it may be stated that with the measuring gauges at a distance of 50 feet from a 300-pound detonating charge of amatol placed about 50 feet below the surface, the corresponding time-pressure curve showed that the maximum pressure, 0.80 ton per square inch, was reached almost instantaneously; that the pressure fell to one-quarter of its maximum value in 1/1,000 of a second, and practically faded away after 5/1,000 of a second. In the course of the investigation it was shown that the pressure waves are reflected from the water surface as waves of tension. The effect at any given point in the neighborhood of the explosion is therefore due to the superposition of a direct pressure wave from the charge and a reflected tension wave from the surface: both travel with the velocity of sound (4,900 feet per second) and the tension wave follows the pressure wave after an interval determined simply by the difference in the length of the direct and reflected paths from the charge to the point in question.

It has been found that the pressure from a large charge is more intense and more sustained than that from a small one, the two being connected by the following rule. If one charge has twice the linear dimensions of another (eight times the weight) the maximum pressure at a given distance from the larger charge will be the same as at half the distance from the small charge and will be twice as sustained, *i.e.*, will take twice as long in falling to any given fraction of the maximum.

When explosions of gunpowder, for example, were investigated, it was found that the pressure rose much more gradually than when charges of amatol or T.N.T. were used. With this explosive very low maximum pressures were obtained and the corresponding pressure waves were considerably prolonged.

Another method of investigating such pressures, which was suggested by Sir J. J. Thomson and applied by Mr. D. A. Keys, consists in the employment of the phenomenon long known to scientists, namely, that certain crystals become charged with electricity when subjected to pressure. The amount of the charge produced is proportional to the pressure applied to the crystals, so by having a suitable arrangement for measuring this charge and its variation with time, a complete record of the variation of pressure with time is obtained by placing the crystal detector at any given distance from the exploding charge. Since the duration of the wave in passing over the crystal or engulfing a submarine is only a few thousands of a second and the pressure generated may be of the order of half a ton

or more per square inch, one can readily imagine the difficult nature of the problem in hand. But by making use of the inertia of a beam of cathode ray particles and employing the fact that they carry negative charges and are deflected by electrostatic and magnetic fields, it has been possible to obtain records of the variation of such pressures with the time. The electrons affect a photographic plate, *i.e.*, they leave an impression on the plate where they strike it. This additional fact has made it possible to determine the change in pressure of the wave from the instant the charge is fired and at as small intervals as we please afterwards. Changes which take place in 1/100,000 of a second have been recorded by this means. Already the method has revealed a number of facts about the nature of the pressure wave produced by exploding charges and the importance of such results in the laying out of minefields and the employment of depth charges can hardly be over-estimated.

(*h*) *Sound Ranging.*—In the course of our investigations of the characteristics of pressure waves generated by the explosion of charges in the sea, it was found that when a hydrophone was used to pick up the waves a good record could be obtained by the explosion of a number 9 detonator at least 2 miles away. The explosions due to charges of 2 pounds of T.N.T. have been recorded at 14 miles, and might have been recorded at far greater distances judging from the strength of the signals received. The explosions of 300-pound depth charges have been recorded up to 200 miles, and it is probable that with charges of moderate amount explosions occurring as far away as 500 miles can be readily recorded. Based on these results a system of sound ranging under water was developed. Four hydrophones were laid out 5 miles apart along a base line in deep water a mile or two from the shore, and in addition two pilot hydrophones were placed along a line at right angle to the base line, the one 5 miles out and the other at twice that distance. Cables were laid from the hydrophones to a recording instrument situated in a shore station. Four of these stations were installed at different places along the east coast of the British Isles, and other stations are now in progress of installation. With such sound-ranging systems the shock of distant explosions occurring under water affect the various hydrophones in turn, and as time intervals can be read to two or three thousandths of a second with the apparatus now in use, it is possible to measure with accuracy the time intervals between the times of arrival of a sound wave at the different hydrophones. With the measurements of these time intervals it is a simple matter to deduce the position of the point at which the explosion setting up the wave is located. Up to 50 miles the location of an explosion under water can be determined to within a few hundred yards by a single station, but for accuracy the co-operation of two stations would be necessary to locate explosions at greater distances. Within operable ranges a ship can be given its position by sound ranging more accurately than by directional wireless or by any other known method. Explosions of mines or torpedoes at any point in the North Sea can easily be located by stations situated in Great Britain. In the war, during the bombardment of the Belgian coast it was a common thing for a monitor to proceed in a fog to a position some miles from the coast and by dropping depth charges have its position accurately determined from stations on the coast of England. So accurately was this done that it was found when the monitor's guns were trained in selected directions objectives several miles inland could be hit with regularity and with a minimum expenditure of ammunition.

(*i*) *Helium.*—Shortly after the commencement of the war it became evident that if helium were available in sufficient quantities to replace

hydrogen in naval and military airships, the loss in life and equipment arising from the use of hydrogen would be enormously lessened. Helium is most suitable as a filling for airship envelopes in that it is non-inflammable and non-explosive, and, if desired, the engines may be placed within the envelope. By its use it is possible to secure additional buoyancy by heating the gas (electrically or otherwise), and this fact might possibly lead to considerable modifications in the technique of airship maneuvers and navigation. The loss of gas from diffusion through its envelope is less with helium than with hydrogen, but on the other hand the lifting power of helium is about 10 per cent less than that of hydrogen.

Proposals had been frequently put forward by scientists in the British Empire and in enemy countries regarding the development of supplies of helium for airship purposes, but the first attempt to give practical effect to these proposals was initiated by Sir Richard Threlfall, who received strong support from the Admiralty through the Board of Invention and Research.

It was known that supplies of natural gas containing helium, in varying amounts, existed in America, and it became evident from the preliminary investigations made by Sir Richard Threlfall and from calculations submitted by him as to the cost of production, transportation, etc., that there was substantial ground for believing that helium could be obained in large quantities at a cost which would not be prohibitive.

The writer was invited by the Board of Invention and Research in 1915 to determine the helium content of the supplies of natural gas within the Empire, to carry out a series of experiments on a semi-commercial scale with the helium supplies available, and also to work out all technical details in connection with the large-scale production of helium and the large-scale purification of such supplies as might be delivered, and become contaminated with air in service.

In the course of these investigations which were carried out, with the co-operation of L'Air Liquide Company, it was found that large supplies of helium were available in Canada, which could be produced at a cost of about 1*s*. per cubic foot.

In the spring of 1917, when the United States of America had decided to enter the war on the side of the Allies, and after the investigations referred to above were well under way, proposals were made to the Navy and Air Board and to the National Research Council of the United States of America to co-operate by developing the supplies of helium available in the United States. The authorities cited agreed to co-operate with vigor in supporting these proposals, and large orders were at once placed by them with the Air Reduction Company and the Linde Company for plant, equipment, cylinders, etc. The Bureau of Mines also co-operated by taking steps to develop a new type of rectifying and purifying machine. By July, 1918, the production of helium in moderate quantities was accomplished, and from that time forward the possibility of securing large supplies of helium was assured.

Concurrently all practical details of the construction of helium-borne airships and of the navigation of this type of craft were developed by the Airship Production Section of the Navy. At the same time, under the direction of the writer, plans were prepared and steps were taken to erect and equip a station for purifying the helium which might become contaminated in service.

Experimental investigations were also initiated with the object of developing all possible technical and scientific uses of helium. In particular, balance, spectroscopic and electrical methods of testing the purity of the gas were worked out, studies on the permeability of balloon fabrics

to hydrogen and helium were commenced, and experiments were begun to exploit the use of helium in gas-filled incandescent lamps; gas-filled arc lamps, and in thermionic amplifying valves. The equipment provided for the purification of contaminated helium in large quantities supplied the major portion of the apparatus required to liquefy helium, and arrangements were therefore made to produce this gas in a liquid form for the purpose of carrying out low-temperature researches.

The advances actually made at the time the armistice was signed warrants the opinion that by the present time, had the work projected been completed, supplies of helium at the rate of 2,000,000 cubic feet a month would have been produced within the Empire and the United States at a low cost, helium-filled aircraft would have been in service, and great progress would have been made in exploiting the technical and scientific uses of this gas.

Before the war a proposal to utilize helium as a filling for airships would have been viewed, even by most scientists, as impossible, but thanks to the enterprise, enthusiasm and initiative of the Navy, backed by imagination, a suggestion—at one time considered to be chimerical—has today become a realization.

DEFENSIVE MEASURES.

From time to time publicity has been given to the steps taken by the British Navy in co-operation with the Navy of the United States of America to close to the passage of submarines such sea areas as the northern portion of the North Sea and the Straits of Dover.

At the time of the signing of the armistice, this stupendous task was well advanced. The material used consisted largely of ordinary contact mines, which were used in vast numbers and at the expenditure of enormous labor and capital.

It can be stated now, however, that concurrently with the installation of this system of defense, other systems of dealing with these and similar areas were worked out, which involved the use of more subtle mechanisms in quantities which were vastly smaller in amount.

Today it is scientifically possible and practicable with a comparatively small amount of material to close effectively to the passage of submarines by either automatic mechanisms or by controlled ones such sea areas as the Bristol Channel, the water stretch between the Mull of Cantyre and the north coast of Ireland, the Straits of Dover, the sea area between Belgium and Denmark, the Cattegat and Skager Rack, and the greater portion of the North Sea between the Orkneys and Norway.

This will serve to show, in a measure, the part science has been able to play during the war. Had the knowledge we now possess been available at the opening of the war, we should have been spared much inconvenience, suffering and anxiety, for the submarine menace, at one time threatening and uncomfortably dangerous, would never have been able to materialize.

APPLICATIONS OF SCIENCE UNDER PEACE CONDITIONS.

Under peace conditions many important technical systems and devices brought forward during the war will find immediate application as aids to navigation.

By means of directional wireless systems, ships or aircraft in the English Channel, the North Sea, or in the eastern and western portions of the North Atlantic, can be given their positions when prevented from getting it by the existence of fogs or unfavorable weather. By means of sound ranging it is possible to fix the positions of light vessels, buoys which indicate channels and obstructions, such as sunken ships. Ships steaming in

fog up the Channel or approaching the shores of Nova Scotia, Newfoundland or Labrador, can be given their positions with accuracy for ranges up to as much as 500 miles. Seaplanes and aircraft in distress in the neighborhood of the British Isles or near the coasts of America can call for help and be located when wireless gear becomes inoperable by simply dropping depth charges.

In hydrographic work generally sound ranging will be of the greatest service for surveys can be made and investigations of seabeds carried out in fogs as well as in fair weather without the delays which have been experienced in the past. The positions of the localities being investigated can always be determined in a few minutes when once a sound-ranging station has been established on some shore within reach.

By Leader gear laid in such areas as the River St. Lawrence, the entrance to the Thames or to Halifax Harbor, the Straits of Dover, etc., in and out lanes of traffic can be organized which can be maintained with ease in fogs. The Echo methods to which reference has been made can be used for founding, for locating icebergs, surface vessels and rock-bound coasts in a fog as well as for locating submarines.

Helium which was originally produced as a filling for airships can be utilized for the production of illuminating agents and for providing a means of investigating the fundamental properties of matter at the lowest temperatures attainable by man. Developments in internal-combustion engines, in electric drive, and in the fuel values of new materials which were to be corporated in the Navy, will be of enormous value to the mercantile marine in the future.

Advances made in wireless telegraphy and telephony and in secret signalling by specific types of radiation for war purposes, will also prove of great service under peace conditions by providing us with novel, efficient, and less costly methods of communication.

PROPOSED SCIENTIFIC ESTABLISHMENTS FOR THE FUTURE.

With a view to developing and extending the scientific results which were obtained under stress of war, the Admiralty has recently put forward proposals for the permanent establishment of a Department of Research and Experiment within the Navy.

Plans have been formulated for the erection of a Centrals Research Institution for the investigation of first principles and for carrying on researches of a fundamental and pioneer character. Steps have already been taken to organize a sea experimental station and to provide buildings and equipment for an engineering laboratory, a wireless and signal school, and a torpedo and mining school in place of the "Vernon." It is believed that these institutions will prove of great value in developing not only means of increasing the efficiency of the Navy, but in providing aids to navigation for our mercantile marine.

The initial expenditure for buildings and equipment will be large, but it seems evident that an ample financial return will in a short time be obtained for the nation from profits accruing from a lowering of the rates of insurance and from a reduction in the cost of transportation. If we could by the use of such aids to navigation as have been referred to above prevent two or three wrecks per year, or lower the time of passage between Great Britain and Canada on the average by one day per voyage per ship through the fog-covered areas in the neighborhood of Newfoundland, sufficient funds would be saved in a year or two to cover the whole cost of the expenditure on scientific and experimental establishments and on the prosecution of the researches and investigations foreshadowed. —"Engineering."

OIL-FUEL INSTALLATION IN PASSENGER STEAMSHIPS.

New Board of Trade Instructions.

In view of the increasing use of oil fuel for boilers, the Board of Trade has issued for the information and guidance of surveyors the following instructions relating to oil-fuel installations on passenger steamers. Excepting the minimum flash point of the oil, which has been reduced from 185 degrees F. to 175 degrees F., the instructions represent, in a general form, the practice which has been followed by the Department in dealing with the numerous individual cases which have been submitted to them.

Flash Point.—The flash point of the oil-fuel should be 175 degrees F., or upwards, as determined by Abel's close test. With each supply of oil taken on board, a written guarantee that the flash point is not below 175 degrees F., signed by a responsible official of the firm supplying the oil, should be furnished. As opportunity offers, the chief engineer should make check tests of the flash point of each supply taken on board, in order that he may have accurate knowledge of the class of oil he is using.

Storage Tanks, Etc.—Oil fuel of the above description may be carried in cellular double-bottoms under engine and boiler compartments, and under ordinary holds; also in peak tanks, deep tanks, and bunkers of approved construction, particulars of which should, in the first place, be submitted for consideration. Provision should be made for the expansion of the fuel in tanks. If the storage and settling tanks are to be constructed to the requirements of either Lloyd's Register, the British Corporation, or Bureau Veritas Registry, a copy of the detailed plans approved by the classification society should be forwarded to the Principal Ship Surveyor for consideration. In other cases, fully detailed plans of the proposed construction, including riveting, should be submitted for approval before the work is taken in hand. Double-bottom compartments used for oil-fuel storage are to be fitted with center divisions. In other storage tanks, suitable wash plates are to be fitted. Where oil fuel is carried in wing spaces on each side of the ship, these spaces must be connected by means of suitable pipes or ducts fitted as low down in the ship as practicable, in order to comply with the requirements of paragraph 61 of the "Instructions Relating to the Construction of Passenger Steamships."

Suitable air and filling pipes are to be provided. The open ends of the air pipes should be led to a situation above deck where no danger will be incurred from the discharge of oil vapor therefrom when the tanks and bunkers are being filled, and each should be furnished with a wire-gauze diaphragm which can readily be removed for cleansing or renewal.

Every oil fuel tank is to be fitted with a sounding pipe led to a suitable position above the crown of the tank. Where air or sounding pipes pass through the cargo holds, arrangements must be made for effectually protecting them from being damaged. No short sounding pipes should be passed either in tunnels or elsewhere.

Provision should be made to prevent over-filling the tanks or bunkers, and gutter ways and coffer dams should be fitted where required. If fresh water is stored in a compartment adjacent to an oil tank, a coffer dam is to be fitted or other effective means taken to prevent the water from being contaminated. If double bottoms under cargo holds are used for the storage of oil fuel, efficient means must be provided by wells or gutter ways to prevent leakage from any oil fuel compartment coming in contact with the cargo, and to ensure that such leakage shall have free drainage into the limbers or wells.

In steamships trading in climates where the cold may cause the oil to become viscous, heating coils should be fitted near the open ends of the

suction pipes in the storage tanks, to render the oil sufficiently fluid to flow freely through the pipes.

Tests.—Every bunker or storage tank is to be tested by filling with water to a head at least 1 foot above the highest point to which oil has access in pipes, hatchways, or elsewhere. In new steamships, the double bottom is, however, to be tested with a head of water up to the bulkhead deck, as required by paragraph 78 of the "Instructions Relating to the Construction of Passenger Steamships."

Settling Tanks.—If the oil is heated in the settling tanks, in order to facilitate the separation of the oil from the water, open drains should not be allowed unless satisfactory precautions are taken that the oil shall not be heated to a temperature at which oil vapor may be given off. Preferably, the drainage should discharge directly overboard, by gravity, or it should be pumped overboard.

Insulation, Etc.—The boilers should be suitably lagged. The clearance space between the boilers and the sides of the storage tanks or bunkers in which oil fuel is carried should be adequate for the free circulation of air necessary to keep the temperature of the stored oil well below the flash point; and bunkers which overhang the boilers in close proximity thereto should be fitted with shield plates.

Funnel Dampers.—Funnel dampers should not, as a rule, be fitted, but, if fitted, they must be provided with a suitable device whereby they may be securely locked in the fully open position. On ships fitted out for the use of oil only, dampers should not be fitted.

The smoke-box doors should be shielded, well fitting, and kept shut.

Pumps.—The pumps for the oil fuel system should be entirely separate from the feed, bilge, and ballast pumps and connections, and should be provided with efficient escape valves which must be in close circuit, *i.e.*, discharging to the suction side of the pumps.

Pipes.—The oil pressure pipes should be solid drawn, and those for conveying heated oil should be placed in sight above the platform in well-lighted parts of the stokehold or engine room. The coupling flanges of the pressure and other oil pipes should be machined, and thin material should be used for the joints so that the flanges may be, practically, metal to metal.

The pipes should be tested to at least double the working pressure.

Cocks and Valves.—All oil fuel suction pipes from storage or service tanks placed above the double-bottom should be furnished with cocks or valves secured to the tanks and so geared that they may be shut off from the deck above or from a compartment other than the one in which they are situated, as well as from the latter compartment. If the discharge pipes are not connected to the tanks near the top, they should likewise be provided with cocks or valves similarly operated, or else non-return valves should be fitted.

Cocks or valves should also be interposed between the pumps and the suction pipes, in order that the pipes may be shut off when the pumps are opened out for overhauling.

It is desirable that no gauge glasses be fitted either to the storage tanks or to the service tanks; but, if suitably protected, gauges of approved special design, or others having flat glasses of substantial thickness and cocks whereby they may be shut off in the same manner as required above for the suction cocks on these tanks may be allowed.

Heaters.—If steam is used for heating the oil fuel, the exhaust drain should discharge the water of condensation into a tank where it can be seen whether or not it is free from oil.

Lighting.—No exposed lights should be placed in any space where oil

vapor might accumulate, such as the pump room or stokehold, which should be illuminated by means of an electrical installation, preferably on the two-wire, two-conductor system, when the electrical pressure exceeds 110 volts, and no switches or fuses should be placed in any such space. The electric lamps should be protected by wire guards if considered necessary.

Electric torches should be provided for examining the bilges, pipes, etc., for leakage, and self-contained, battery-fed lamps, similar to those used in fiery mines, may be used in dangerous places. Portable lamps supplied with current through flexible cables should not be permitted.

Precautions Against Fire.—In addition to the usual water service and hose and pipe conductors, one of which might be provided with a nozzle having perforations in it to spray the discharge, pipes perforated for the emission of steam into the gutter-ways or lower parts of the boiler room should be supplied, the control of steam supply being outside the boiler room. The boiler room should contain a receptacle for holding about 5 cwt. of sand, and suitable scoops for distributing the sand should be provided, together with chemical fluid extincteurs for use in the boiler room, pump room, or engine room.

Contiguous Passenger or Crew Spaces.—Neither passengers nor crew should be berthed or accommodated in a space adjoining an oil bunker and separated therefrom only by the containing bulkhead of the bunker. No objection need be raised, however, to passenger or crew spaces situated on a deck forming the crown of an oil fuel space, provided that the deck is constructed thoroughly oil tight, that there are no manholes or hatchways to the oil spaces situated within the quarters, that the flooring is laid with a non-inflammable and non-magnesite composition at least 1½ inches thick, and that the spaces are especially well ventilated.

Outfit.—Thermometers and an apparatus for determining the flash point of the oil fuel should be supplied; also extinguishers, in the form of tubes with one end closed, for the torches used in igniting the oil spray at the burners.

Printed or typed instructions regarding the working of the system should be furnished for the guidance of the engineers; also a plan, suitably mounted, of the oil-piping arrangements.

Miscellaneous.—The escape of oil fuel heated to or above the flash point is most dangerous, and may result in an explosion or a fire should a naked light come into contact with the highly inflammable gas which is evolved.

Ample ventilation should be provided in engine, boiler and pump rooms, and also in all compartments adjacent to the storage tanks.

After lighting the burners, the torches should on no account be thrown away before carefully extinguishing by means of the appliances provided for the purpose.

Savealls should be provided under the pumps, heaters, and strainers where necessary, to catch oil which may be spilled when any cover or door is removed; and, likewise, at the furnace mouths when not protected by fronts which would intercept oil having escaped from the burners or becoming extinguished.

There should be no woodwork in the stokehold or compartment containing the settling tanks, and no wood or other combustible matter should be allowed to accumulate therein or in the vicinity of the fuel tanks.

Cleanliness Essential to Safety.—No oil should be allowed to accumulate in the bilges or gutter ways or on the tank tops. These parts should be washed out with a hose, having a conductor, at least twice a day, or oftener if required, and the wells should then be pumped dry. In order

that the gutter ways may be readily accessible for inspection and cleaning, it is desirable that the hold platform should be kept clear of all bulkhead plating.

Before any tank or bunker which has contained oil fuel is entered for inspection or repair, the oil should be entirely removed, and care should be taken that all oil vapor is also removed by steaming or by efficiently ventilating. Satisfactory tests of the atmosphere in the tanks or bunkers should be made to ensure safety against explosion before inspection or work in them is begun.

General.—If any difficulty arises in securing the adoption of the arrangements indicated in these instructions, the Surveyor should refer the matter for consideration by the Engineer-Surveyor-in-Chief.—"Steamship."

THE MEASUREMENT OF CONDENSER PERFORMANCE.

The extent to which the steam turbine is used at the present time for the propulsion of ships makes the problem of efficient condenser design one of the greatest importance. It is generally recognized by marine engineers that the best results from a steam turbine can only be obtained when the steam is allowed to expand to the lowest possible pressure before passing through the exhaust to the condenser, and hence, on turbine ships, it is customary to instal condensers which are capable of producing a higher vacuum than is the case on ships fitted with reciprocating engines. The production of an extra half inch in the condenser vacuum will result in a large increase in the power developed by the turbine, and it has become the practice to run the condensing plant on turbine ships so that the vacuum approaches to within about 1 inch of barometric height, or, on an average, to about 29 inches of mercury, while on vessels fitted with reciprocating engines, a 26-inch or 27-inch vacuum was generally considered as representing the limit above which it was not economical to proceed. It is not necessary at the present juncture to explain why it is that the turbine installation is always associated with a higher vacuum in the condenser, but rather to consider the various means which are adopted for producing this result, and to try to establish some basis of comparison in order to determine to what extent each method may be considered as an economical procedure.

Considering the condensing plant as comprising the condenser itself with merely its circulating pump and air pump, an increase in the vacuum can be obtained by increasing the size of the condenser or by fitting a larger air pump when dealing with a given quantity of exhaust steam. Merely increasing the size of the circulating pump tends to lower the temperature of the condensate with corresponding loss of efficiency of the boiler; although in dealing with a larger condenser a correspondingly larger circulating pump will of course be required. Hence, to produce a higher vacuum, we shall be faced with a larger expenditure of steam in both the air pump and the circulating pump, and the question arises as to how far these increases can be met before the saving in consumption in the main turbine is balanced by the increased consumption in the condenser auxiliaries. In addition to the factor of steam consumption, there is also the question of the increased weight and cost of the plant to be set against the saving in the cost of running it, and this is a point which must not be lost sight of when dealing with the increase in the size of auxiliary machines for any particular purpose.

Although no definite data are available as to when the simple condensing arrangement referred to above ceases to be economical, it is realized that in the production of high vacua some additional means must be adopted.

There are many systems of producing a high vacuum which have been tried in conjunction with turbine installations on board ship, and all of them achieve a certain measure of success. In one system we have what is termed an augmenter condenser by means of which the vacuum obtained in the main condenser is still further increased. In another system we have two air pumps, one connected to the bottom of the condenser and one to the top, and known respectively as the wet and dry air pumps. Yet, again, we have systems in which rotary pumps are employed in preference to reciprocating pumps, while there are many devices in which the exhausting effects of jets of either steam or water are employed in order to reduce the pressure in the main condenser. All these systems, as we have already said, succeed more or less in their object of producing a high vacuum, but it is very difficult to say at what expense in steam consumption this vacuum is obtained, and in any case the marine superintendent who is desirous of forming an opinion as to the relative merits of the different systems with which he is confronted, is at a loss to find a common basis from which the systems may be compared.

In measuring condenser performance there are so many variable quantities to be taken into account, that it is a difficult matter to find a common denominator in terms of which the performance of any plant may be expressed. Neglecting the question of the prime cost of the different systems, we might determine the number of pounds of steam condensed per pound of steam employed in the auxiliaries; but it must be remembered that these figures will only be of value when the pressure of the steam used in the auxiliaries is considered. Again, the temperature of the circulating water will materially affect the result, while the pressure, and hence the temperature at which the exhaust steam is condensed, must also be considered. But it is apparent that merely to say that a condensing system is capable of giving a vacuum of so many inches, or so much per cent of barometer, is not sufficient unless some indication is also given of the expense in steam consumption, and in other terms, with which this vacuum is obtained. The growing importance of the steam turbine for merchant ship propulsion and the consequent demand for condensing plant capable of yielding high vacua, renders it almost imperative that some means of expressing the efficiency of the performance of condensing apparatus should be devised. The above ideas have only been framed with a view to pointing out some of the difficulties of the question and to suggest a means whereby comparative figures might be obtained. But while the problem is one of great difficulty, it is one of great importance. Many systems of condensing are available. Many more will doubtless appear as the demand increases, and the superintending engineer will be called upon to make a choice as to which system is the most economical from all points of view. In helping him to make that choice, some basis of comparison is clearly desirable.—"Shipbuilding and Shipping Record."

WATERTIGHT DOORS IN MERCHANT SHIPS.

It is now possible to analyze experience during the great test of war on the question of watertight doors in ships. The controversy usually centers around the need or otherwise for doors in watertight bulkheads, and the difficulty has been expressed by saying that the question is one of the safety of the ship *versus* the convenience of the departments working the ship. It was abundantly demonstrated during the war that watertight doors when left open are a positive source of danger. The majority of vessels lost were sunk by the explosion of a mine or torpedo, and it was found that any bulkhead in the vicinity of the explosion was usually distorted to

such an extent that it was impossible to close the door. Actually it was found that if the door was open at the time of the explosion there was generally, with few exceptions, no time to shut it even if it had been in working order. It has often been proved that watertight doors are open when an unforeseen contingency arises. It has also been shown that it is not possible to rely upon a crew, however well disciplined, to close watertight doors in an emergency. Examples of this occurred during the war, where in many cases a watertight door was far enough from the center of the explosion not to be distorted, but only in isolated instances were such doors shut after the explosion occurred. For example, a large passenger vessel, 500 feet long, was torpedoed without warning and capsized and sank in 5 minutes. The explosion laid open two compartments to the sea, but this would not have been in itself sufficient to sink the ship. Unfortunately the watertight door leading into a third compartment was open at the time, so that actually three compartments were opened to the sea. The large area of waterplane lost on this account was enough to destroy the stability of the ship, making her capsize. Other cases occurred where damage was received in the after part of the ship, destroying a shaft tunnel and thus admitting water to the engine-room through the watertight door at the forward end of the tunnel which was left open.

In the case of a collision a bulkhead between two compartments may be damaged, and it is of the utmost importance that the inrush of water should be confined to the smallest limits. The suggestion was made by the Court of Inquiry into the *Empress of Ireland* disaster that in circumstances of danger it would be desirable to close all watertight doors and port holes below the top of the watertight bulkheads and to keep them closed until the danger was passed. It will be remembered that the *Empress of Ireland* was sunk in collision with a collier; many lives were lost. The advice given by the Court of Inquiry is undoubtedly sound in theory, but in practice—especially in peace time—since the closing of ordinary watertight doors causes such inconvenience in the ship, it is not likely to be taken. If it is logical to leave out watertight doors in a ship because they are dangerous, it is equally as logical to leave out port holes.

During the war, a committee appointed by the Institution of Naval Architects recommended that all existing watertight doors low down in main bulkheads should be secured so that they cannot be opened, and that if watertight doors be necessary they should be fitted high up in the bulkhead. This was considered by the committee to be particularly important when bunker coal is carried forward of the boiler-room bulkhead, as a watertight door through which coal is being trimmed cannot be depended upon. In the ordinary way this would mean lifting the coal in the reserve bunker over the top of the boiler-room bulkheads, or through the watertight door openings cut high up in the bulkhead, thus necessitating extra work for the crew of the ship and an increase in their number.

In spite of all that has been said for and against watertight doors, the fact remains that watertight doors are still fitted, the evidence pointing to the convenience of working the ship outweighing the possible danger brought about by fitting the doors. That this is so seems fairly obvious, since the necessity for closing a door in an emergency may come once in, say, ten years, but the desirability of having it there occurs several times a day.

Access must be provided to coal bunkers, and although every effort should be made in designing a ship to arrange that the bunker bulkheads are non-watertight, this is by no means always possible, particularly under the Subdivision Rules. Then, again, there must be access between the boiler rooms and from them to the engine-room. Shafting requires

periodical attention and watertight doors must be provided in the shaft tunnel for this purpose. It has been suggested that if watertight doors are dispensed with in the machinery spaces, access can be obtained to each watertight compartment from the top deck. This would mean increasing the number of engineer officers in a ship and adding to the wages bill. In addition, access trunks would have to be fitted in the 'tween decks or the engineers and greasers allowed in passenger accommodation. It is true that watertight doors are greatly cut down in warships, at least in the main watertight bulkheads, but in these vessels the number on the engine-room staff is not important, and in any case there are no passengers to consider. Economical considerations are relatively of little importance.

Even in warships, where efficiency is the first consideration, the number of watertight doors fitted is enormous. Some twenty years ago there was a sustained discussion with regard to this very question, the principal participants being the late Sir William White and the late Lord Charles Beresford. The former was in the position of having to put in watertight doors to meet the wishes of the senior naval officers controlling the design of warships. Lord Charles Beresford was anxious to reduce these, and while it is true he made some very drastic suggestions with regard to watertight doors low down in the main machinery bulkheads, he only proposed to do away with 19 out of a total of 208 in the battleship *Magnificent*. Since then some of his recommendations have been carried out, but even today in a first-class battleship there are nearly 200 watertight doors.

In passenger ships, watertight doors are generally necessary in one 'tween deck. It is curious in this connection, that the latest Subdivision Rules have actually added to the number of watertight doors required in a ship. It was usual before these rules came into force to stop the watertight bulkheads at a deck below the weather deck. Now, to get satisfactory subdivision, they must often be carried up to this latter deck. In the shelter-deck type of vessel, for instance, the main transverse bulkheads were only carried to the upper deck, and in consequence the passenger accommodation could be arranged in the shelter 'tween decks without introducing watertight doors in that space. Under the new Subdivision Rules it is frequently found necessary to carry the watertight bulkheads up to the shelter deck. In practice, as already indicated, it is necessary often to move from one compartment to another in the shelter 'tween decks, so that the bulkheads must be pierced and watertight doors fitted. These doors must be left open when the vessel is at sea. Watertight doors in a 'tween deck may be hinged or sliding, depending on the height of the deck above the water line. The danger of watertight doors in passenger 'tween decks being left open has often been demonstrated. When a damaged vessel has sunk to such a level as to bring the water into this 'tween deck over the damaged compartments, the water has promptly spread throughout the ship.

If there should be doubt still as to the need for watertight doors it should be removed by reference to the report of the Bulkheads Committee on the Subdivision of Merchant Ships. This report says, "The number of openings in watertight bulkheads shall be reduced to the minimum compatible with the design and proper working of the ship."

To summarize the position; watertight doors are objectionable principally because they are kept open and because in the ordinary way they require a great amount of effort and a good expenditure of time to open and close them. The necessity is for a means of opening and closing watertight doors easily. To do this the door must be operated by power. With such doors the convenience of the departments is met without endangering the safety of the ship. The principle of having watertight

doors always closed can be applied to all those which are operated by power in the lower part of the ship, as, for instance, those in the machinery spaces. This disposes of the objection that after a ship has received damage it is not possible to close the door, since it is already closed. On the other hand, many prefer to leave the doors in the machinery spaces open.

We have remarked that it is quite hopeless to rely upon direct human action locally, so that the necessity for power operation controlled from some central position becomes apparent. The great advantage of a central control from the bridge is that this control is under the ruling brains of the ship, *i.e.*, either the captain or the first officer, men who have been trained to act quickly and with presence of mind in an emergency and who, moreover, will actually see danger approaching in 99 cases out of 100, and will be able to throw over their door control lever before the actual moment of impact, thus forestalling the danger of distorted door frames to which reference has already been made. With regard to doors in passenger 'tween decks which cannot be kept closed without a great deal of inconvenience, it is safe to say that the 'tween deck bulkheads would not usually be distorted by an ordinary peace time accident unless the bulkhead was completely destroyed. In consequence it is possible to operate these doors, but that they should be closed quickly goes without saying, and what has been said for power-operated doors, centrally controlled, applies.

With regard to watertight doors in bunker bulkheads, the great objection to them is that they are likely to become partially blocked with coal. Here, again, the power-operated door has a distinct advantage in that the power can be made great enough to make the door cut through the coal. Nearly all the prominent steamship lines have fitted their passenger vessels with a centrally-controlled, power-operated watertight door system, and reports show that during the war it has been more difficult to sink these, than ships fitted with the ordinary watertight doors. With the power-operated doors there is often a certain amount of nervousness at first, amongst the men, with regard to their use, but in practice it is found that this is overcome when the men see that the local control is in their own hands and is quite sufficient to act against the central control, although the latter is made to close the door after the local action is taken off, this being usually done automatically. A power system of watertight doors does cost extra money, but this can be looked upon as an insurance premium.

The International Convention Rules for watertight doors, in vessels carrying more than 200 passengers, make it necessary to have, either doors which close by their own weight or by power pressure, and in any case operated from the bridge. Actually it is not often possible, in practice, to make all the watertight doors in the machinery spaces slide vertically so that they will close by their own weight. The result is that power-operated doors must be fitted, so that the Convention Rules do, in effect, require a power-operated, centrally-controlled system of watertight doors in passenger steamers.

In a passenger vessel, therefore, the choice remains between solid bulkheads and centrally-controlled, power-operated doors, and the advantages of doors are so obvious as compared with the inconvenient system of unpierced bulkheads that, in these days of high wages and short working hours, it follows that the moderate expense of installing an efficient power system would quickly be exceeded by the wages bill where solid bulkheads were fitted. It appears certain that all liners will in future have their bulkheads pierced for watertight doors, and that such doors will be centrally controlled and operated by power.—"Engineering."

GERMAN COASTAL SUBMARINE MINE-LAYERS.

(C Class).

By Dr. Eng. Franz Werner, of the Naval Board.

When in the autumn of 1914 Flanders had fallen into our hands, the necessity arose for having a mobile coast defense as well as a stationary one. For this purpose the war administration requested the speedy construction of small submarines which were to carry torpedoes and mines. The submarine commission fulfilled this order by designing two boats, one solely for torpedoes, the so-called *B-I* type (compare "Schiffbau" XX No. 18, pages 485 to 496), the other solely for mine-laying, the *C* type. Work was begun on the first of the C type (designated in this article as *C-I*) in November, 1914. The Vulcan Works at Hamburg was commissioned to build ten and the Weser Stock Company five.

C-I TYPE.

The plans had been made by the construction department of the submarine commission. In the following only the characteristic distinctions will be taken up, as the properties peculiar to the coastal submarine, and the conditions governing its construction have been explained in the above-mentioned article about the *B* type, which also holds good for the mine-layer. The principal measurements are shown in Tables I to V. Table I (combination drawing) shows the outward form. The boats are especially remarkable for three peculiarities which exercised a lasting influence on the design.

FORM.

Especially noticeable with regard to the form is the ungainly cylindrical shape of the fore part which reminds one of a Zeppelin airship. There were two reasons which led to the choice of this peculiar shape. The first one was the space required for the mine-tubes in which the mines were stowed. The second was the necessity of transporting the boats to Flanders by rail. This led to the reduction of the maximum diameter to a size which would be suitable to the dimensions permitted by railroad cars, tunnels, bridges, etc. The shape of the special railroad cars necessitated the outlines of the fore part of the vessel.

CONNING TOWER.

The second characteristic of the boats is the high conning tower. All the foreign submarines of the same size were furnished with observation hoods, and not with real conning towers, consequently the commander's post is only slightly above the water when traveling on the surface. The submarine commission knew very well that these vessels would only be able to maintain themselves if they could travel on the surface for considerable distances. This, however, necessitates a sufficient sea-worthiness. The latter can only be insured if the conning tower is tall enough to give protection from the sea to commander and helmsman. The natural outcome of this was the furnishing of the commander's post with a pressure-resisting substructure tall enough for at least one man to stand upright in.

TABLE 1

CHIEF MEASUREMENTS AND QUALITIES

	UC-I	UC-II	UC-III
Length over all (m)	34.0	51.85	56.10
Maximum beam (m)	3.15	5.20	5.55
Depth of keel with maximum oil supply on board (m)	3.04	3.645	3.765
Height of keel (m)	.3	.56	.56
Height of keel (lower section) with upper edge O.D. measured for ½ length (m)	5.2	4.6	4.7
Height to top of periscope (m)	6.3	7.6	7.7
Surface displacement with (a) normal oil supply (cbm)	177.0	417.0	480.0
(b) maximum " " "	177.0	433.0	491.0
Displacement submerged (a) without free flooding spaces (cbm)	192.5	508.6	559.27
(b) total form displacement (cbm)	225.0	550.0	716.0
Capacity of ballast tanks including additional bunker (cbm)	14.6	82.02	91.0
Capacity of trimming tanks (cbm)	5.0	20.3	24.1
Capacity of ordinary bunker (cbm)	2.5	46.6	63.6
Cap. of additional bunker (cbm)		16.6	12.5
Capacity of lubricating oil tanks (cbm)	0.52	5.67	4.6
Surface engines H.P.	90	500-600	600-650
Maximum surface speed, knots per hour	6.5	11-12	11-12
Surface cruising radius, nautical miles	800 at 5.5 kn.	10000 at 7 kn.	8000 at 8 Kn.
Under water engines H.P.	138.0	460.0	600.0
Number of cells and type of accumulator battery	2 x 56 18 MAS	2 x 62 26 MAS	2 x 62 32 MAS
Capacity of the accumulator battery for 10 hours in ampere hours	2700.0	4800.0	5900.0
Maximum speed submerged knots per hour	5.0	7.0	6.6
Maximum cruising radius, submerged, at 4 knots per hr., nautical miles		55.0	56.0
At 3 kts. per hr., nautical miles		92.0	95.0
Torpedo equipment (maximum number carried)	- - - -	2 tubes for'd 1 tube aft 7 torpedoes 50 cm	2amidships 1 aft 7 torpedoes 50 cm
Mines	12.0	18	14
Guns	1 M.G.	1-8.8 cm (100) 1 M.G.	1-10.5 cm (170) 1 M.G.
Number of crew	15	28	32
Admissible depth for diving (m)	50	50	75
Time of diving to a depth of 9 m. (sec)	30	40	45
Total cost of construction in marks	700,000	1,700,000	3,000,000

TABLE II

STABILITY MEASUREMENTS

	UC-I	UC-II	UC-III
1. Stability on surface			
F (Center of displacement) above lower edge of pressure hull (m)	1.45	1.722	1.865
M (Metacentric width) above F (m)	.07	.537	.613
G (Center of gravity) above lower edge of pressure hull (m)	1.27	1.804	1.949
Metacentric height MG	0.25	0.455	0.529
2. Stability under water for γ = 1.0			
F above lower edge of pressure hull	1.59	1.986	2.043
G above lower edge of pressure hull	1.24	1.804	1.909
FG (length & Breadth)	0.35	0.182	0.134

TABLE III

DATA CONCERNING PRESSURE HULLS AND OTHER IMPORTANT PARTS

	UC-I	UC-II	UC-III
Length of pressure hulls (m)	29.62	40.40	43.0
Max. Dia. " " (m)	3.15	3.652	3.652
Thickness of plates of pressure hulls (mm)	10 Mine space 11 on side 14 above and below	11 Mine space 11 on side 12 above 13 below	11 Mine space 11 on side 12 above 13 below
Frame profile (mm)	L 130 x 65 x 8 (amidships) L 150 x 70 x 10 (mine space)	L 150 x 70 x 10	L 150 x 70 x 8.5
Distance between frames (mm)	750	800	800
Total weight of pressure hull (Sides, front.. (kg)	35470	55800	62655
Total volume of pressure hull (cbm)	183.5	314	338.6
Weight of pressure hull per cbm of vol. kg/cbm	19.35	17.8	18.5
Measurements of conning Tower (mmø)	1250	1350	1350
Thickness of top of conning tower (mm)	15	12	12
Plate thickness of outer hull (mm)	-	3.5-4	5
Frame profile of outer hull (mm)	-	∠ 60x40x5	∠ 60x40x5 50x40x5
Distance between frames of outer hull (mm)	-	4-500	600
Plate thickness of spaces open to the sea (mm)	1-2	2	2

TABLE IV

	UC-I	UC-II	UC-III
Main rudder (sqm)	1 x 1.75	2 x 2.15	2 x 2.15
Forward rudder (sqm)	2 x 1.0	2 x 1.75	2 x 1.75
After rudder (sqm)	1 x 2.0	2 x 1.6	2 x 1.6

ARRANGEMENT FOR THE MINES.

The third peculiarity is the arrangement for the mines. In the fore part of the boats are six slanting tubes which pass completely through the pressure hull so that, when the vessel is submerged, the tubes are entirely filled with water. In each tube two mines are stowed one above the other. See drawing 5. The cams reach under the projections *b* of the mine anchors and thus hold them firmly in place in the tube. The cams may be screwed into the openings *c* of the mine tube from the interior of the boat whereby the mine is deprived of its support and drops downwards. The mine had to undergo special improvements in order to adapt it to the dropping contrivance. Our mines up to that time were not adapted to this arrangement. But it was necessary to avoid changing the lead ignition cap which had already successfully stood the test of use. The ignition cap of the mine was now considerably endangered by the walls of the tube particularly during the fall so that special contrivances had to

TABLE V.

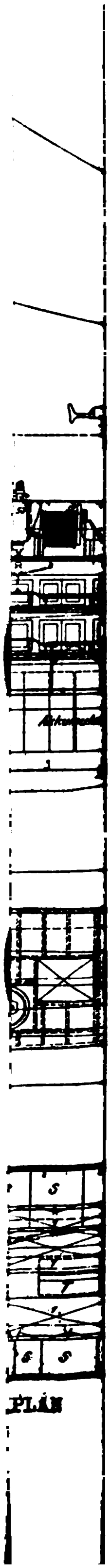
WEIGHTS

	UC-I		UC-II		UC-III	
	t	%	t	%	t	%
Hulls	81.25	46.0	182.1	45.00	205.0	42.8
Engines	40.50	22.9	111.3	26.00	144.6	30
Torpedo equipment	-	-	14.7	3.5	14.1	3.0
Gun equipment	-	-	4.2	1.0	10.7	2.3
Mine equipment	14.0	7.9	14.67	3.5	12.1	2.5
Fixtures			6.8	1.6	5.5	1.1
Materials and lub. oil			5.8	1.4	6.5	1.4
Crew. provisions	15.5	8.7				
Water			11.7	2.75	12.2	2.6
Fuel, normal			41.0	9.25	56.8	11.6
Ballast and reserves	25.75	14.5	24.7	6.0	13.3	2.7
	177.0	100.0	417.0	100.0	480.0	100.0

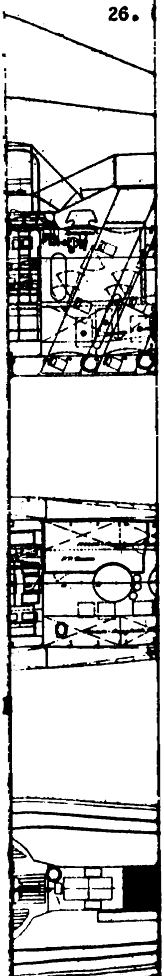
DISPLACEMENT

	UC-I (cbm)	UC-II (cbm)	UC-III (cbm)
Volume of pressure hull	169.0	313.98	338.5
Outer hull, volume (Pressure proof cells and additions)	9.0	57.4	78.9
Displacing fuel		46.6	63.6
Total submerged displacement without ballast cells	178.0	418.0	481.0
Residue	1.0	1.0	1.0

be made to ensure the safe exit of the mines from the U-boats. For this purpose hinged guide-rails were attached to the mine-anchor in such a way as to give the mine the correct direction while leaving the tube, thus protecting the lead ignition caps from contact with the walls of the tube or with the vessel. (See drawing 5.) After the mine has dropped down and has settled on the ocean-bottom, these guide-rails fold back and lie on the ocean bottom like grippers, in this way permitting the mine to float freely upwards and the anchor cable to unwind without hindrance.


PLAN

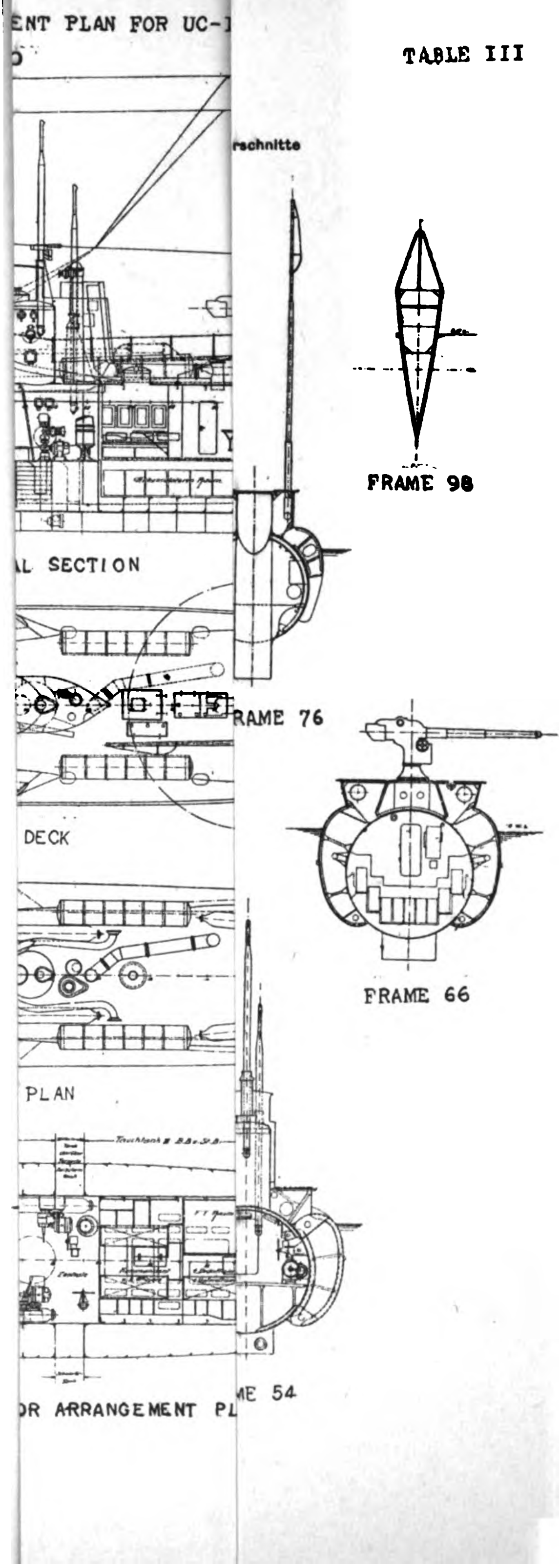
14.
15.
16.
17.
18.
19.
20.
21.
22.
23.
24.
25.
26.



I

47.
OUT
48.
49.
50.
51.
52.

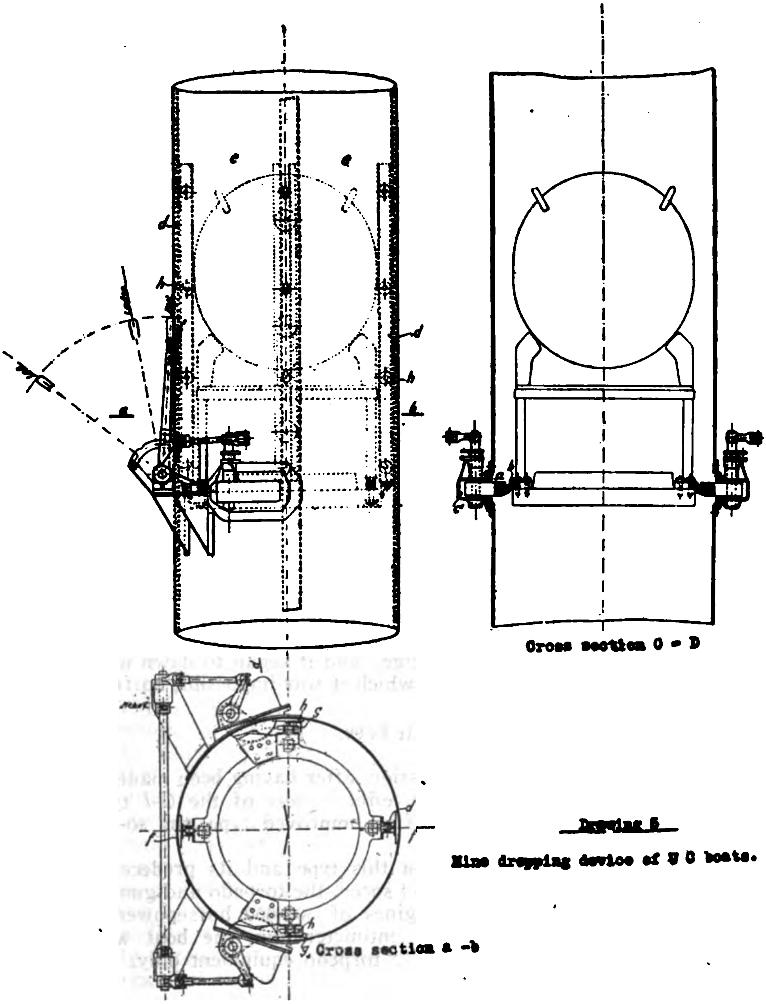
TABLE III



ENT PLAN FOR UC-I
rschnitte
AL SECTION
RAME 76
DECK
PLAN
OR ARRANGEMENT PL
ME 54
FRAME 98
FRAME 66

THE MINE-TUBES.

It was necessary to incline the mine-tubes so that the mines might be released while the boat was in motion. The degree of inclination was determined by experiments with models. The experiments were executed on the basis of the highest attainable water-velocity although the mines are



Drawing 5

Mine dropping device of U C boats.

usually released at considerably lower rates of speed. It was, however, observed that an inclination which was greater than the respective speed of the vessel's demands is not detrimental to the falling of the mine.

The mines are guided during their exit by flat iron rails *f*, two attached lengthwise and two crosswise in the angles *g* arranged in the cross-section of the tube. For this purpose the above-mentioned guiding-rails of the mines are furnished with guiding-rollers *h*. The axes of the latter are alternately radial and tangential to the mine.

PRESSURE HULL.

The pressure resistance of the pressure hull required especial care in the region of the mine-tubes as the perforating of the pressure hull by the tubes annuls the crosswise resistance. The position of the tubes to one another was determined in such a way that between each one a circular frame could be executed the entire length without a break. As on the other hand the intervening space was too large in relation to the surface resistance of the pressure hull, intermediate frames were inserted with their ends resting against the mine-tubes. The pressure thus occurring was taken up by powerful journal-bearings on the perforations at the top and bottom of the pressure hull. In order to increase the resistance of the journals, the topmost and the lowest strakes of the pressure hull were made thicker than those at the sides.

EFFECTIVENESS.

Briefly, it may be asserted of this type of boat that it fulfilled the required stipulations. In the scant six months which elapsed from the commencement of the design in November, 1914, to the delivery in May, 1915, nothing more technically perfect could have been evolved. The fact must be taken into consideration that for the Weser Yards (Bremen) and the Vulcan Works (Hamburg) U-boat construction was something new and gave them some hard nuts to crack.

Very soon after the boats had been put into active service it became evident what a valuable weapon they might be in the hand of a clever leader. However, the demand for stronger offensive properties in the new boats that were to be constructed was only natural.

At the time when the *C-I* boats were designed we were still counting on a short war, the autumn of 1915 at the longest; but when they were completed the state of affairs had changed and it began to dawn upon us that we were facing a war the end of which it was impossible to foresee.

C-II TYPE.

Therefore, the submarine commission, after having been made acquainted with the judgments passed on the effectiveness of the *C-I* type and its weaknesses, took up the design of an improved type, the so-called *C-II* type. (See table I.)

The essential difference between this type and its predecessor lay in its far superior offensive properties: speed, the torpedo and gun equipment. By the insertion of two main engines of 500-600 horsepower combined, three torpedo-tubes and one 8.8 centimeter gun, the boat was able to attack as effectively as U-boats with torpedo equipment only.

TORPEDO EQUIPMENT.

As the mines occupied the entire space in the pressure hull the only remaining position for the torpedo-tubes was on the outside of the boat.

In order to be able to retain accuracy of aim it was decided to choose pressure-proof, water-tight outside tubes and to dispense with the use of the so-called grating or despatch tube used by the French and the Russians.

The place adjacent to those portions of the mine-tubes which projected above the pressure hull turned out to be the suitable location. Here the tubes could be placed without disturbing the outer lines parallel to the axis of the boat to any considerable degree. It was also possible to place them in such a way that they could be used when above the surface of the water or when submerged. This is not always easily accomplished, as the arc of fire of the torpedo must be kept clear in front of the torpedo-tube, therefore the boat must be shaped accordingly.

THE HULLS.

For the shipbuilder the essential difference between *C-I* and *C-II* type lies in the construction of the hull. The former is built with a single hull, the latter with a double hull.

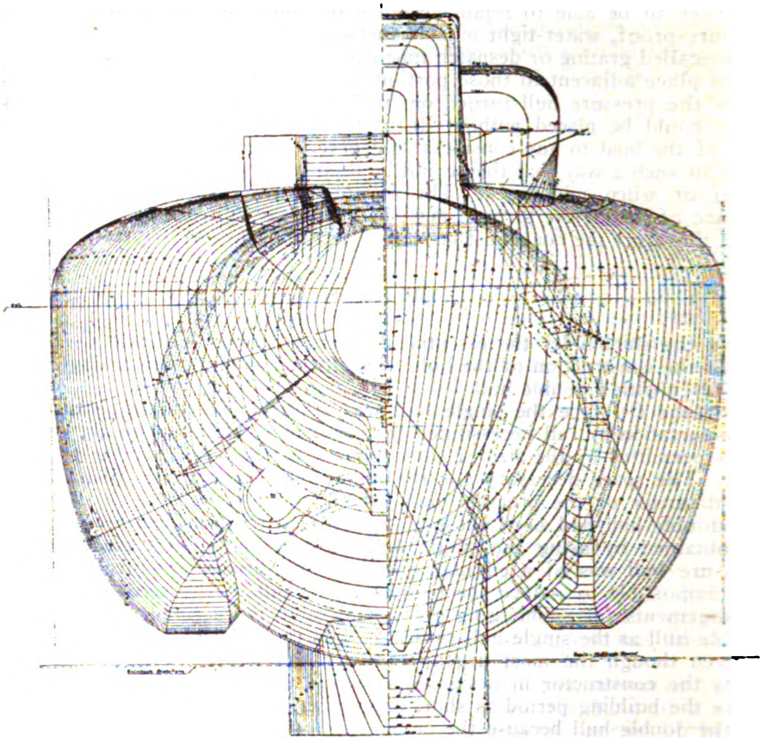
For the *C-I* type the single hull was a matter of course because the transportation by rail necessitated the complete exploitation of the dimensions permitted by the size of cars, tunnels, bridges, etc., for the pressure hull. The space necessary for the mine-tubes forbade any reduction in the diameter of the pressure hull, which would have been necessary if the double hull had been chosen. Speedy construction also was only to be obtained by using the single hull because in the first place, as the pressure hull was of a circular cylindrical or circular-spherical shape, it was impossible to reduce the templet-work, and in the second place the arrangements for submerging are considerably more simple than for the double hull as the single hull possesses only two interior ballast tanks.

Even though the most urgent necessity had been supplied by the *C-I* boats the constructor in designing the *C-II* type was still constrained to make the building period as short as possible. In spite of this he decided on the double hull because the better form attained in this way promised a more favorable adaptation of the engine arrangement. Besides this the double hulled boat has better sea-going properties and more stability for travel. This point was of the utmost importance as the boats were to work at least two weeks at a stretch on the high seas. Another reason for the choice was the much greater radius of action which it is possible to give to a boat of this type.

The objection will be offered that the new single hulled boats, such as those of the American Electric Boat Company, possessed good lines for surface travel. But the reason for this is that the thick outer plates of the single pressure hull have double curved surfaces which are difficult to make, whereas the pressure hull of the double hulled boats has plates having a single curve only.

Thus the gain in time of construction on account of simple submerging arrangement would have been equalized by the construction of the more difficult form of the plates; and the double hulled boats would still have the larger radius of action in their favor.

Especially noteworthy in regard to the *C-II* type is the way the lines have been carried out in relation to the mine tubes which project from the lower portion of the ship thus explaining the relatively broad keel. (See reproduction 8.) For the rest they have throughout the appearance of a normal surface vessel.



Reproduction 8 C II type Construction plan.

EFFICIENCY.

Boats of this type have stood the test of service well; sea-worthiness, submerging qualities, its action when submerged and the manoeuvering capabilities gave perfect satisfaction. The obtained speed of 11½ to 12 knots was, in view of the strong offensive qualities, relatively high; the radius of action in spite of the small displacement was so extensive that the boats could be employed all round the British Isles, in the Atlantic as far as the coast of Morocco, and could be sent from Kiel to Pola without renewing the fuel supply.

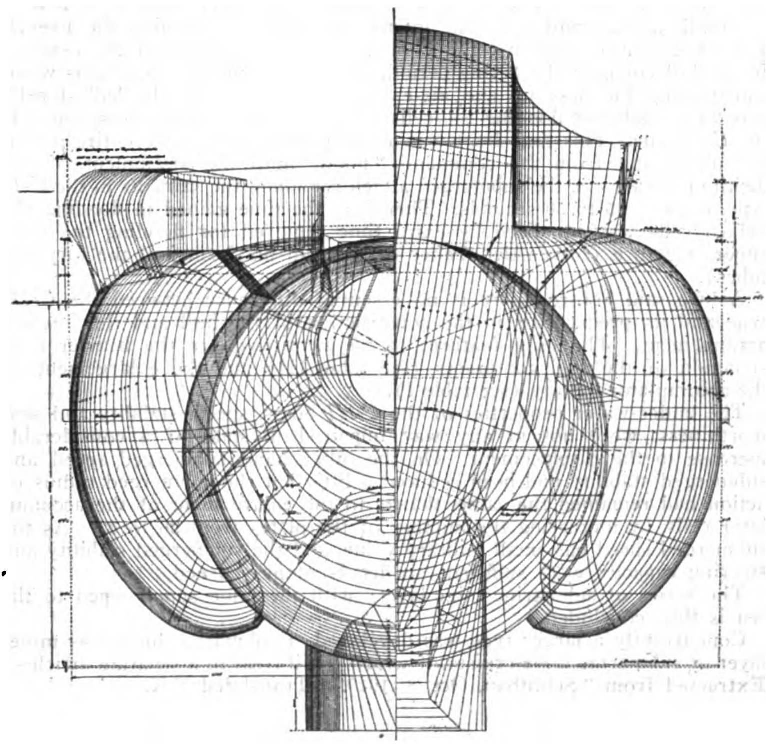
But they possessed two features which caused the chiefs to instigate a change in the construction of the *U-I* when at a later date, mine layers were ordered in considerable numbers. These features were the wet bridge which was a consequence of the speed combined with the high forecastle and the position of the torpedo-tubes at the side; and the easily flooded gun platform which was caused by the well occurring between forecastle and conning tower. Both of these were military disadvantages not to be underestimated. They rendered the sighting and recognizing of the adversary very much more difficult and impaired accuracy when aiming the weapon.

A submarine more than any other war-vessel is never a thing of perfection but will always remain a compromise, with defects of more or less importance. Therefore such defects as those mentioned above, although in themselves small, stand out so prominently for the reason that they interfere with the daily life of the crew and they are therefore often subject to more severe and lasting criticism than more serious and fundamental defects such as lack of speed, stability, etc.

C-III TYPE.

When, therefore, after a time of less necessity for mine-layers, there arose again in 1917 the demand for a great number, the Submarine Commission decided to design a slightly different type, but which only deviated outwardly from the *C-II* type, the interior remaining practically unchanged. The details of the plans were prepared at the wharf of Blohm and Voss and the Danzig Imperial wharves, who alone constructed or prepared the boats of this type with the exception of three which were built by the Weser yards; Blohm and Voss having the construction of 89 to their credit.

The new form shown in table III was the result of the effort to avoid the defects mentioned. (See reproduction 9.) It is characterized by the



Reproduction 9 C III type Construction plan.

gently sloping strake of the forecastle to the gun platform. The position of the gun itself is, however, more elevated and the torpedo-tubes which were formerly outside in the bows and were said to cause the spray were placed much farther back, so that the mouth pieces were abreast of the conning tower.

It was evident that these compromises also possessed disadvantages. Even while making the design, the constructor found it most convenient for the torpedo tubes to be no longer placed parallel with the lengthwise axis of the hull, thus ensuring a free field for the torpedo projectile, but at an angle which deviated several degrees from the above. The result was a very unfavorable shape for submerged navigation. The new forecastle, which took in the gun platform, increased the submerged displacement considerably. Above all, the torpedo tube greatly enlarged the maximum midship frame when submerged. (See reproduction 9.) The result of the trial trips was a not inconsiderable decrease of the highest speed.

The new position of the torpedo tubes had done away with the disagreeable spraying of the bridge, but in place of this, when in a sea of four points, brightly shining lines of foam were formed at the caps of the torpedo tubes, thus increasing the visibility of the vessel when in motion. This again was a military disadvantage.

In another direction also the U-boats of the *C-III* type were unable to fulfill all expectations. The intense vigilance in guarding the French and English coasts against the U-boats constantly increased the demand for a shortening of the diving period. The time limit set as a basis when constructing the flooding and air-expelling apparatus of the ballast cells was 20 seconds for flooding the cells. This was ten seconds less than for the *C-II* type. This time limit was actually attained, but the entire period of submerging from the command "flood" until the attainment of the depth necessary for the submerged attack remained the same as in the *C-II* type, namely, 45 to 50 seconds. Therefore the time gained in flooding the ballast cells was equalized by extra time required for flooding the open super structure (forecastle, shelter of the gun platform and the torpedo-tubes).

As we said before, there is no essential difference between the *C-III* type and its predecessor in the construction of the hull and the interior arrangement. The only features to be mentioned are the heavier construction of the fuel containers (see table III) and the enlargement of the accumulator battery (see table I).

The under-water properties of the *C-III* boats, speed, stability and seaworthiness, were not in any way impaired in spite of a considerable increase in the displacement. On the other hand submerged speed and submerged stability had been reduced a little but the submerged radius of action had remained the same, thanks to the enlargement of the accumulator cells, by increasing the number of the plates from 26 to 32. As the submerged speed has small military value, and the submerged stability and steering capacity were sufficient no disadvantages ensued.

The series of submarine mine-layers with the mine supply open to the sea is thus concluded.

Concurrently a larger type was developed, a submarine high seas mine-layer or submarine mine-cruiser of which I will treat in a separate article.—Extracted from "Schiffbau, Oct. 8, 1919." Translated E.K.

OBITUARY.

THEODORE COOPER.

Old officers, and particularly the older Engineers of the Navy, and the students at the Naval Academy from 1866 to 1869, will remember Theodore Cooper, who was, in his day, one of the brightest and most genial of the Engineers.

He was born in Coopers Plains, N. Y., eighty-one years ago and died in New York City during August last.

He was graduated at the Van Rensalaer Institute of Technology in the class of 1858, and entered the Navy on the 24th of December, 1861, as a Third Assistant Engineer. He served on board the *Chocura;* in the many battles in the Pamunky River; at Yorktown; Sabine Pass; and Calcasieu. After the war had ended he made a cruise on board the *Nyack,* in the Pacific, after which he became an instructor at the Naval Academy.

He was promoted to Second Assistant in 1863 and First Assistant in 1866, and resigned in 1872. He liked the service; liked the life and he resigned reluctantly; promotion being so slow, and every reorganization of the Navy reduced the engineers more than other corps, which discouraged Mr. Cooper. He accepted a lucrative position, offered by Capt. Eades, in charge of the construction of the St. Louis bridge: After that was finished he became Superintendent for both the Delaware and the Keystone Bridge Companies.

"When the first elevated railroads were built in New York City, Mr. Cooper was the assistant engineer in charge of construction. He was one of the five experts named by the President to figure on the Hudson River Bridge span and was consulting engineer for the Quebec Bridge and the New York Public Library. And the time the first cantilever bridge across

the St. Lawrence, at Quebec, collapsed and nearly 100 men lost their lives, it is said that the accident might have been avoided had a telegram sent by Mr. Cooper been received and heeded," says the *New York Sun* of 26th August, 1919.

Mr. Cooper was a member of the board of experts on the Manhattan Bridge plan in 1903. He was a lifelong member of the American Society of Civil Engineers, of the Mechanical Engineers, the Loyal Legion and N. J. Historic Society.

He was a robust man, rarely ever complaining, and his death came after a very brief illness of pneumonia, at his residence, 353 West 57th Street, New York. He was never married, which, he so often said, was a reason for remaining in the Navy. His life was a busy one, which, he said, brought content. He was of a genial disposition and had many friends.

G. W. B.

ASSOCIATION NOTES.

Council Meeting October 2, 1919.

Due to detachment from duty in Washington, Lieutenant Commander F. W. Sterling, U. S. N., Retired, and Captain of Engineers M. W. Torbet, U. S. C. G., tendered their resignations as Secretary-Treasurer and Member of Council, respectively, effective October 1, 1919. Commander Joseph S. Evans, U. S. N., was appointed Secretary-Treasurer and Captain of Engineers W. M. Prall, U. S. C. G., Member of Council for the remainder of the year 1919.

The matter of holding a banquet during 1919 was considered, and it was decided that conditions have not yet become sufficiently settled to warrant such action.

ANNUAL MEETING.

The annual meeting of the Society for the nomination of officers for the year 1920, and for such other business as might be presented was held on Tuesday, October 7, 1919, pursuant to our By-Laws. The following were nominated:

For *President:*
Rear Admiral C. W. Dyson, U. S. N.

For *Secretary-Treasurer:*
Commander J. S. Evans, U. S. N.

For *Member of Council:*
Rear Admiral B. C. Bryan, U. S. N.
Captain Robert Stocker, C. C., U. S. N.
Commander S. C. Hooper, U. S. N.
Commander F. J. Cleary, U. S. N.
Commander Kenneth Whiting, U. S. N.
Commander H. T. Dyer, U. S. N.
Captain of Engineers W. M. Prall, U. S C. G.

It was voted to place the matter of holding a banquet during 1920 before the membership and provision has been made for voting on this question on the ballot for officers for 1920.

The Secretary-Treasurer desires to renew his request that members promptly notify him of changes in address.

The following members and associates have joined the Society since the publication of the last Journal:

MEMBERS.

Ames, Eugene, ex-Commissioned Officer, U. S. N.
Gary, C. B., Ensign, U. S. N.
Russell, George L., Lieut., U. S. N.

ASSOCIATES.

Abrams, Lawrence B., Washington, D. C.
Ball, Howard J., Washington, D. C.
Dias, Peter, Lieut. U. S. N. R. F.
Dodge, Owen, Colorado Springs, Colo.
Edmundson, W. A., Washington, D. C.
Harter, Isaac, Supt., Babcock & Wilcox Co., Bayonne, N. J.
Kessler, Raines, New York City.
London, William J. A., President, Steam Motors Co., Springfield, Mass.
Muir, Reginald L., New York City.
Williams, K. D., Washington, D. C.

NEWPORT NEWS SHIPBUILDING and DRY DOCK COMPANY.

WORKS LOCATED AT **NEWPORT NEWS, VA.,** ON HAMPTON ROADS.

Equipped with

THREE LARGE BASIN DRY DOCKS

of the following dimensions:

	DOCK No. 1.	DOCK No. 2.	DOCK No. 3.
Length on top, ...	610 feet	827 feet	587 feet
Width on top, ...	130 "	162 "	147 "
Width on bottom, . . .	50 "	80 "	62 "
Draught of water over sill, .	25 "	30 "	24 "

Shops are equipped with modern machinery capable of doing the largest work required in ship construction.

TOOLS DRIVEN BY ELECTRICITY AND COMPRESSED AIR USED IN CONSTRUCTING AND REPAIRING VESSELS.

For further particulars address

H. L. FERGUSON, President, Newport News, Va.,
F. P. PALEN, Vice President, 233 Broadway, New York.

WELDED STEEL STEAM and WATER DRUMS

for WATER-TUBE BOILERS.

Embodying Strength with Lightness and freedom from LEAKAGES Incident to Riveted Structures.

6-WELDED STEEL WATER DRUMS

MORISON SUSPENSION FURNACES.

FOX CORRUGATED FURNACES.

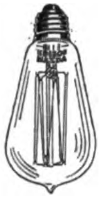
Welded Steel Tanks, Receivers, etc.

The CONTINENTAL IRON WORKS,

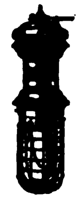
West and Calyer Sts., BOROUGH OF BROOKLYN, N. Y.,

ESTABLISHED 1859
INCORPORATED 1887

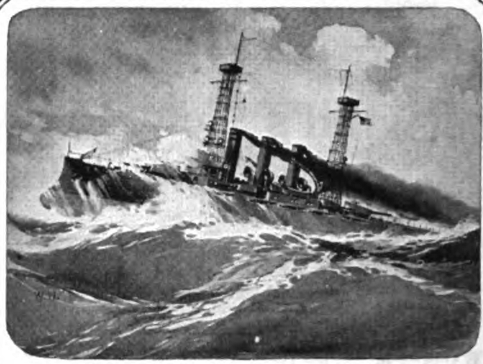
Greenpoint Ferry from East 23d Street, New York.



Mazda Lamp



Marine Arc Lamp



Electric Equipment for the Ship

The General Electric Company is prepared to furnish electrical equipment for marine use from the small electric fan in the state-room and cabin to the powerful steam turbines used for ship propulsion.

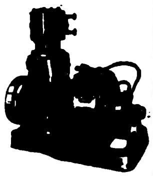
If steam is not available, one of our internal combustion engine generating sets will supply current for motors, lights, heating, cooking and operating of searchlights.

Our marine engineers are thoroughly familiar with the installation of electrical equipment on all types of ships.

5-30

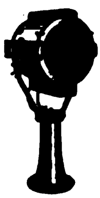
GENERAL ELECTRIC COMPANY

GENERAL OFFICE SCHENECTADY, N.Y.

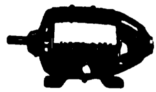


Steam Generating Set

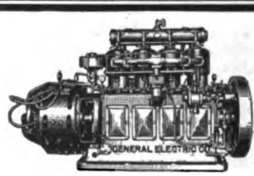
Fan



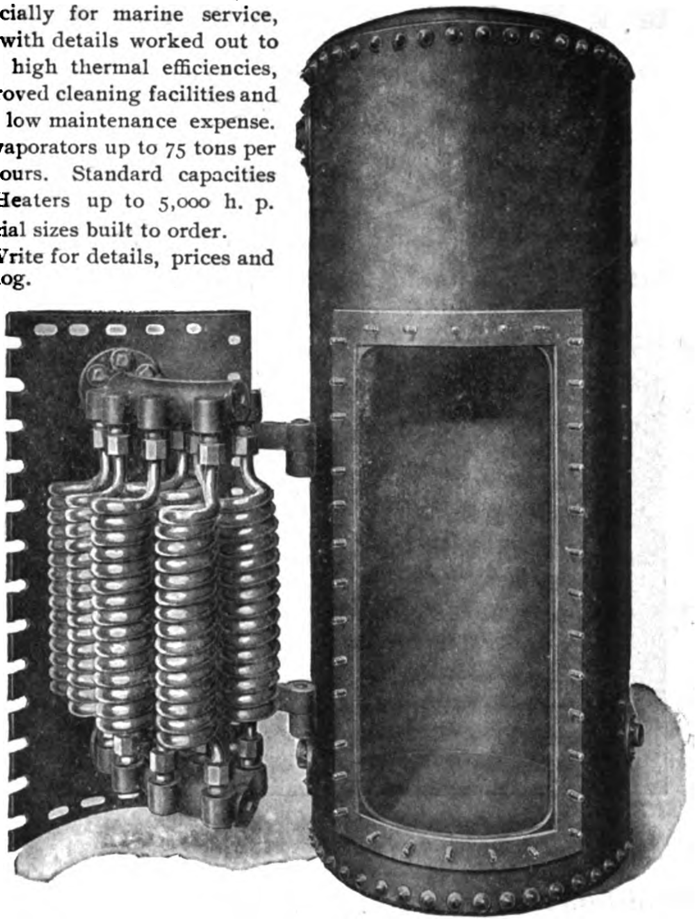
Searchlight



Motor



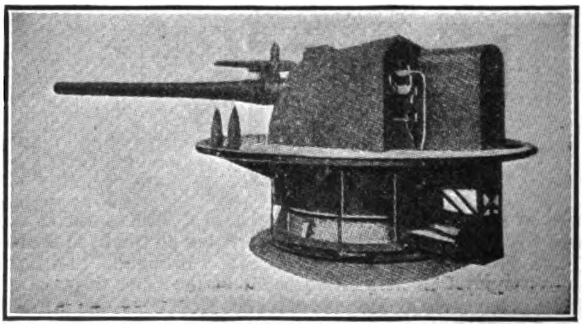
Internal Combustion Generating Set



BETHLEHEM STEEL CO.

SOUTH BETHLEHEM, PA. 25 Victoria Street, LONDON

Naval, Field and Coast Defense GUNS and MOUNTS Armor Turrets Projectiles



Forgings Castings Shafting Rails and Structural Steel

9."2 Coast Defense Gun with Sub-calibre Equipment, on Barbette Mount.

We are continuously manufacturing ORDNANCE MATERIAL for the U. S. Army, U. S. Navy and Foreign Governments

SAVE TIME AND LABOR IN HANDLING

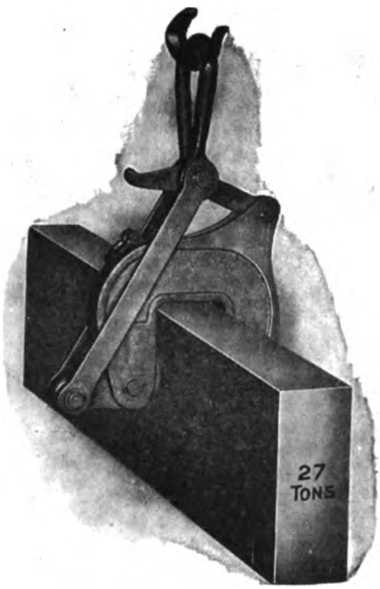
STEEL PLATES

WITH THE

POSITIVE PATENT LIFTING CLAMP

—IN USE AT—

U. S. War Department
Sandy Hook Proving Ground
U. S. Navy Yards
Portsmouth
Boston
Philadelphia
Norfolk
Mare Island
Naval Stations
New Orleans
Pearl Harbor
Hawaii
U. S. Naval Proving Ground
Panama Canal
and Leading Industrial Establishments Throughout the World



British Admiralty Yards
Chatham
Portsmouth
Sheerness
Pembroke
West India Docks
Royal Ordnance Factory
Woolwich
New South Wales Government
French Admiralty
Royal Arsenal
Venezia Government
Ship Yard
Sorel, Quebec
Canada

AND NEARLY EVERY SHIPYARD IN THE UNITED STATES

From 1/2 to 100 Tons Capacity
TYPE "A"

A Clamp with a POSITIVE GRIP
The Heavier the Load, THE FIRMER THE GRIP

WILLIAM E. VOLZ, M. E.

Sole Manufacturer for the U. S.

SALES AGENT

WELDLESS CHAINS, LIMITED

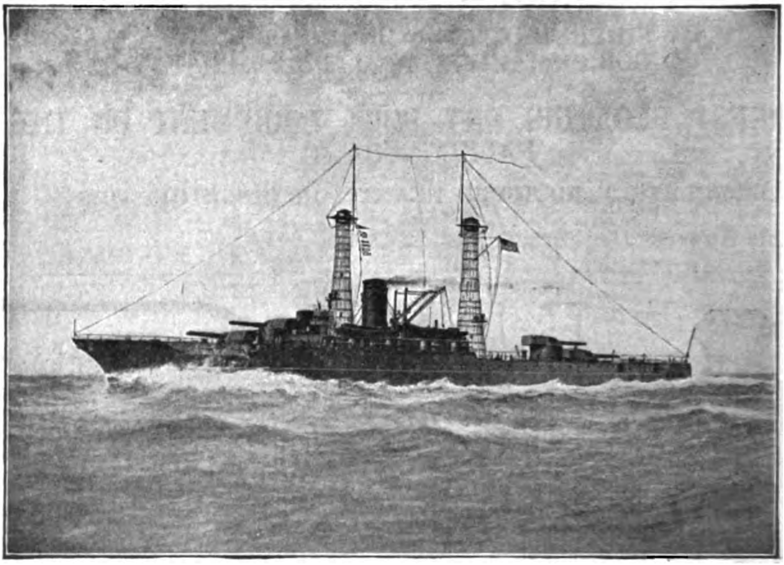
The Strongest Chain in the World

Write for Catalogue

---

6 Church Street - - - - New York

The Slogan of the Cameron—"Character: The Grandest Thing"



CAMERON CENTRIFUGAL PUMPS

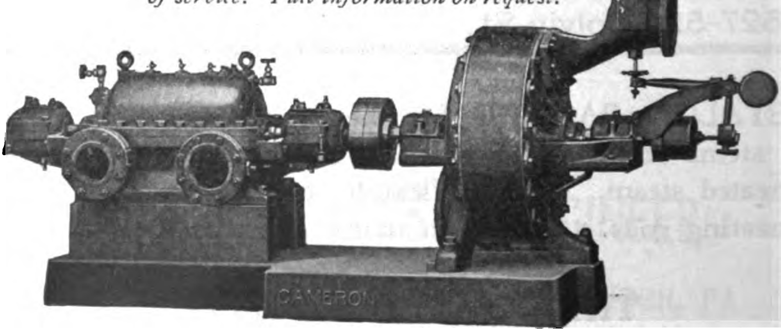
are now in satisfactory service on the following U. S. Naval Vessels:

"CONNECTICUT" "TENNESSEE"
"MINNESOTA" "ARIZONA"

In addition to the above the super-dreadnaught "New Mexico" will have three Cameron Turbine Driven "BT" Centrifugal Pumps, as shown below, for boiler feeding.

The service is particularly severe, as it requires continuous operation at 450 pounds pressure, with a great range in capacity and with high efficiency.

Cameron Centrifugals are built for all classes of service. Full information on request.

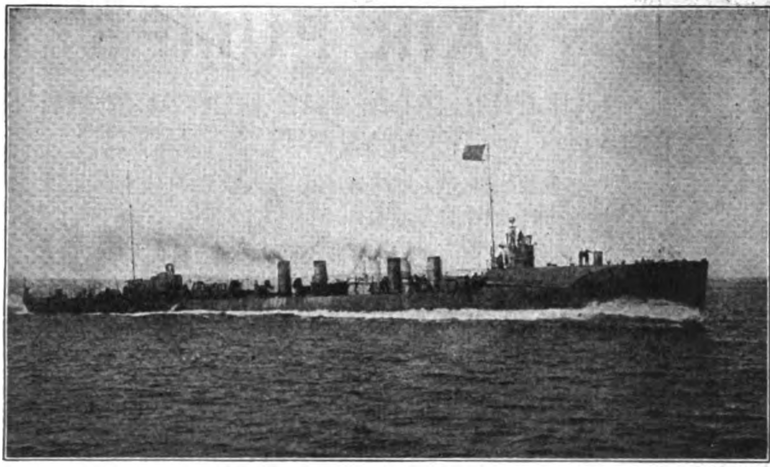


A. S. CAMERON STEAM PUMP WORKS

11 BROADWAY, NEW YORK **Offices the World Over**

BATH IRON WORKS, LTD.

BATH, MAINE.



BUILDERS OF DESTROYERS, FAST YACHTS AND PASSENGER STEAMERS. NORMAND WATER TUBE BOILERS. PARSONS MARINE STEAM TURBINES. TWENTY-FIVE DESTROYERS BUILT OR BUILDING FOR THE UNITED STATES NAVY, ALL HAVING SPEEDS BETWEEN THIRTY KNOTS AND THIRTY-SEVEN KNOTS.

JOHNS-MANVILLE MARINE PRODUCTS

For years we have made a special study of the Naval Engineer's requirements. The results are seen in Johns-Manville Naval Products, known and used wherever ships are built. Our nearest branch will gladly send full information about these Johns-Manville Naval Specialties.

Covers the Continent

Through—

Asbestos

and its allied products

INSULATION

CEMENTS

ROOFINGS

PACKINGS

LININGS

FIRE PREVENTION PRODUCTS

- Packings
- Mastic Flooring
- Asbestos Roofing
- Bitumen-Mastic Solution Enamel and Cement
- Fire Extinguishers
- A-S Refrigerating Machines
- Transite Asbestos Wood
- Navy Brand Fire Felt
- "Noark" Enclosed Approved and Refillable Fuses, Service Switches and Accessories
- Ebony Asbestos Wood
- Steam Traps
- Fibre Conduit
- Asbestos and Magnesia Pipe and Boiler Insulation

WRITE FOR CATALOGS

H. W. JOHNS-MANVILLE CO.

NEW YORK CITY

10 Factories. Branches in 63 Large Cities

RADOJET
AIR PUMPS

THE RADOJET

FULFILL ALL THE REQUIREMENTS OF MARINE SERVICE

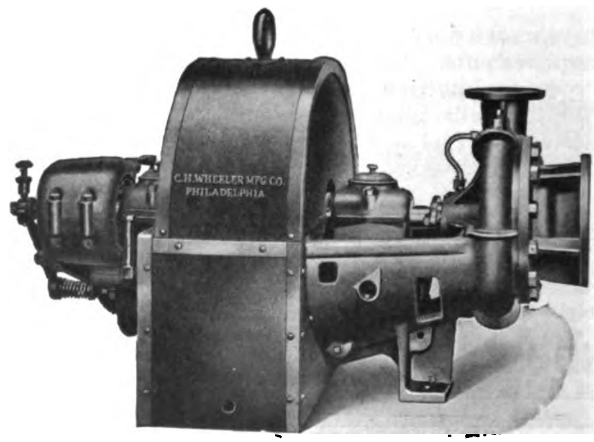
SMALLEST WEIGHT AND SPACE
RELIABLE AND NOISELESS OPERATION
WITHOUT ATTENTION OR REPAIRS
HIGHEST ECONOMY
NO MOVING PARTS
NO LUBRICATION
NO FOUNDATION

C. H. Wheeler Manufacturing Co.

Head Office and Works

18th Street and Lehigh Ave.
PHILADELPHIA, PA.

OVER 1,000,000 H. P. ON ORDER OR IN USE.



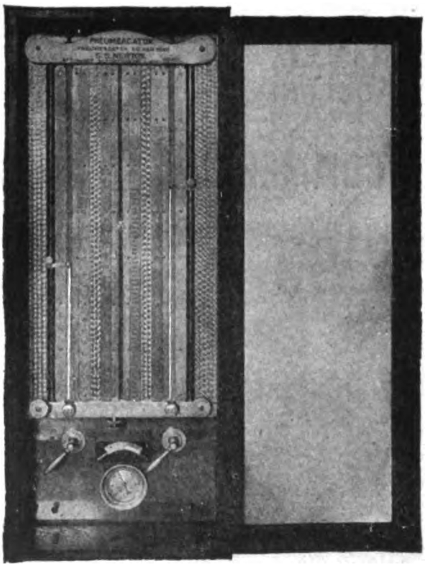
MARINE TYPE CONDENSATE PUMP

PNEUMERCATOR

MODEL D-1—Indicates forward and aft drafts. Registers mean draft and corresponding tons displaced. Shows trim, checks invoices and deliveries, weighs cargoes and bunkers. Other types of this model indicate forward, aft and midship drafts only.

MODEL T-1—Indicates depth and volume or weight of contents of fuel oil, fresh water, ballast tanks or bilges. Checks invoices or withdrawals. Furnishes a perpetual inventory. Other types of this model can be arranged to ring a high or low alarm or both, or to start and stop an electrically driven pump.

Model T-1



Model D-1

PARTIAL LIST OF USERS

The Union Sulphur Co.
New England Coal & Coke Co.
W. R. Grace & Company
Pioneer Steamship Company
Oceanic Steamship Company
Florida East Coast Railway Co.
Mexican Petroleum Company
Warner-Quinlan Asphalt Co.
Ward Line
Luckenbach S. S. Co.
United States Government
U. S. Emergency Fleet Corp.

PNEUMERCATOR COMPANY, INC.

15 PARK ROW
PARK ROW BUILDING NEW YORK, N. Y.

Harry S. Parks 450 N. Tenth St. Philadelphia, Pa.	W. H. Hughes Hippodrome Bldg. Cleveland, Ohio	J. C. H. Ferguson Monadnock Bldg. San Francisco, Cal.	W. S. DePerris Co. 103 Grand Trunk Dock Seattle, Wash.

Kelvin, Bottomley & Baird, Ltd., 16 Cambridge St., Glasgow, Scotland.

The R-P Trade-Mark And What It Means

After fifteen years of engineering and manufacturing experience—years of pioneer work in the development of scientific lubrication—the products of the Richardson-Phenix Company have come to be recognized as standard and Richardson-Phenix lubrication engineers as the final authority on all problems pertaining to scientific lubrication.

In order that all products, and all literature and advertising of this company may be instantly identified, the trade-mark reproduced here has been adopted. This trade-mark is more than a mere label—it is a symbol of a service we alone can render, a service that embraces every branch of scientific lubrication, the correct application, reclamation, and handling of lubricating oils.

THE RICHARDSON-PHENIX CO.

LUBRICATION ENGINEERS AND MANUFACTURERS

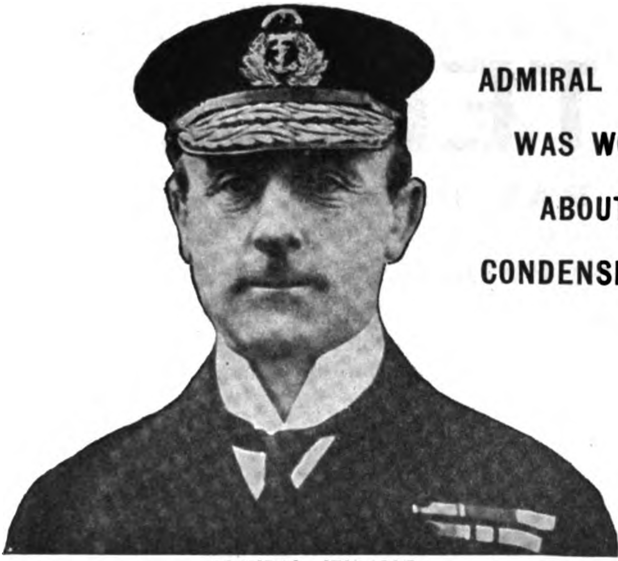
WORKS: 114 RESERVOIR AVE. MILWAUKEE, WIS.

Automatic Equipment for the scientific application, reclamation and handling of Lubricating Oils.
Automatic Oiling and Filtering Systems.
Oiling System Accessories.
Oil Filters and Reclaimers.
Force Feed Mechanical Lubricators.

NoKut Globe and Angle Valves.
NoKut (Scoville Patent) Pump Valves.

Bulletin A-25 fully describes R-P Automatic Oil Circulating and Filtering Systems. It contains some mighty interesting engineering data and is well worth your investigation. Write for your copy. It is waiting.

R
Automatic
Lubrication Equipment
P
SCIENTIFIC LUBRICATION



ADMIRAL JELLICOE

ADMIRAL JELLICOE WAS WORRIED ABOUT HIS CONDENSER TUBES

He seemed to be more anxious over his condensers than over his 11-inch guns. The condenser tubes in many of his ships were in bad condition.

The regular commercial brand of tubing is NOT gooa enougn for condensers. Special painstaking care must be exercised in manufacturing this product to do it RIGHT.

A few years ago we couldn't get the high quality condenser tubing we wanted, so we built a large mill and are now making our own. Wheeler Crescent Brand tubes are drawn and heat treated by EXPERTS IN CONDENSER WORK.

It will pay you to specify CRESCENT BRAND or order direct from our mill.

NO ORDER
TOO LARGE
OR
TOO SMALL

During one month of this year we made and shipped nearly 1,000,000 pounds of condenser tubing—A RECORD.

Shall we put you on our tube department mailing list?

WHEELER CONDENSER & ENGINEERING CO.

CARTERET, N. J.

Manufacturers of:

Condensers, Centrifugal Pumps, Vacuum Pumps, including the patented Wheeler Steam Jet Air Pump, Evaporators, and other condensing equipment for marine service.

38